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ADELPHI-GODDARD EMULSIFIED FUEL PROJECT

Adelphi University  
Garden City, New York

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CENTER FOR ENERGY STUDIES  
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ADELPHI-GODDARD EMULSIFIED FUEL PROJECT

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CENTER FOR ENERGY STUDIES  
ADELPHI UNIVERSITY

GODDARD SPACE FLIGHT CENTER  
NASA CONTRACT #NAS5-23688  
FINAL REPORT

I. Introduction

The NASA project grew out of Adelphi University's prior investigations of the combustion properties of ultrasonically generated emulsified fuels. This was motivated by recent work done at several institutions, including Adelphi, which indicated that water/oil emulsions when used as a fuel, produced significantly less soot than straight fuel oil. In addition, there were reports from installations where the manufacturer of a commercial device had run some spot tests which purported to indicate substantial improvements in boiler efficiency. A degree day comparison with a similar loading period without emulsification, and some spot checks on CO<sub>2</sub> levels and stack temperatures indicated an improvement in the Adelphi boilers. Though these reports were suggestive of an improved boiler efficiency, the tests were by no means scientific and included many different boiler types and fuel oils ranging from No. 2 to No. 6 oil. There are reasons to believe that No. 6 oil, which is ordinarily difficult to burn, would have improved combustion characteristics in emulsion form. However, since the No. 2 oils are easier to use, and in general will not violate environmental restrictions, it was felt that a carefully controlled, scientifically monitored test on Goddard's boilers should be first done with No. 2 oil.

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Our preliminary findings indicated that during combustion, the internal water droplets vaporize, causing mini-explosion of the fuel drops, leading to a much finer atomization and a very thorough mixing of air and fuel. This allows complete combustion with much less air and a reduction in soot production. Less excess air means that less heat is carried out the stack by the exhaust gases. In addition, the reduction of soot keeps the boiler heat transfer surfaces clean, and, therefore, more efficient. Adelphi is also investigating the possibility that the system allows more effective radiative heat transfer from the flame to the boiler tubes. What these improved combustion characteristics mean in a practical sense is that a boiler furnace which ordinarily becomes less efficient with usage, can operate over extended periods of time close to design efficiency. Other data also confirms these findings. Recent tests at the EPA Laboratories in Research Triangle Park, North Carolina, have confirmed a soot reduction of 80-90%<sup>(1)</sup>. Earlier results by Shearer and Tranie in France, by the Battelle Memorial Institute, and at Adelphi University have revealed a dramatic reduction in soot concentrations in boiler furnaces<sup>(2,3,4)</sup>. In the Soviet Union, fuel emulsions have been used extensively since the 1950's in order to obtain improved combustion in boiler furnaces, both in ships and in stationary power plants, and again the importance of soot reduction was noted.

Our Phase I Goddard project was an investigation of the effect of water/oil emulsion on the thermal efficiency of boilers and the reduction of soot in the combustion gases, to verify to what extent real savings can be obtained for No. 2 oil by using emulsions instead of pure oil.



The boiler under study is a water tube steam generator at the Goddard Space Flight Center. The boiler is rated at 35 million BTU's per hour. The boiler has the capability of using three types of fuel No. 2 oil, heavy residual oil, and gas. At the present time, due to the gas shortage, No. 2 oil is the primary fuel. The burner is a steam atomization system. However, when No. 6 fuel oil is used, which is considerably cheaper (approximately 8%), and which has more BTU's per gallon (approximately 10%), the Goddard boilers violate EPA standards due to the production of soot. It is, of course, here where we would expect to have emulsions prove very economical.

The system that is presently used for emulsification is an in-line ultrasonic emulsifier. The emulsion is created just prior to insertion into the boiler. The diagram of the emulsion combustion system with boiler instrumentation is shown in Figure 1.

Before the mixture of water and oil is fed into the emulsifying chamber, it is pre-mixed. The emulsification takes place by irradiating the water/oil mixture with ultrasonic waves of 20 kHz which break the water/oil interface, dispersing tiny drops of water, less than 20 microns, into the oil. The concentrations of water are varied from 10 to 20% by weight. The emulsion is sufficiently stable for burner flowthrough, atomization and combustion. Initial tests at Goddard were carried out with No. 2 fuel oil, more readily available, cleaner and amendable to preliminary testing.

Figure 2 shows the results of the thermal efficiency of boiler #1 for No. 2 oil and for water/oil emulsions. Though there was some slight increase at low fire using emulsions for most of the range of use of this boiler, there was no difference in thermal efficiency between the No. 2 oil and the emulsion.

Figure 3A,B shows the effect of using emulsions on smoke number at several different loads

In Figure 3A when these results are compared with Figure 3B we see that for given excess air the smoke number using emulsions is generally lower. This is what was expected. However, with the steam atomizing system that is used at Goddard, smoke levels are low in general, so that this reduction in smoke number cannot be effectively used to reduce excess air sufficiently with the emulsions to obtain significant increases in thermal efficiency. Though it generally was possible to reduce excess air with the No. 2 oil emulsions, this mostly occurred in the low fired region which is not the normal operating region of the boiler.

As can be seen from Figure 2, the low fire region is one of low thermal efficiency and it is not efficient to run the boiler there for any extended length of time. The conclusion drawn from these tests is that, as far as using emulsions for No. 2 oil on Goddard boilers, there is not a sufficient gain on operating efficiency to warrant their installation.

In our Phase II project we have attacked the problem of burning the heavy oils since it is here that the emulsification is expected to be most beneficial. Our conclusion was that for No. 2 oil with an efficient boiler such as at Goddard, water/oil emulsification was not warranted. From the beginning, we have recognized the desirability of extending these tests to heavy residual oils, e g., No. 6. Some of the potential advantages of using emulsions of the heavier fuels are:

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- a) they require less processing to make from crude oil and are, therefore, cheaper by the gallon, and more plentiful than No. 2 oil. They also have 10% more BTU's per gallon,
- b) environmental standards presently restrict the use of such oils because they generate too much soot. Emulsions provide a proven ability to reduce that soot and permit more widespread use;
- c) more of an efficiency increase is expected for No. 6 oil than for No. 2 oil due to reduction of soot and improved atomization,
- d) they have greater emulsion stability due to naturally occurring surfactants and the fact that No. 6 oil is closer in density to water;
- e) emulsions with soda ash dissolved in the water can also be expected to reduce sulfur dioxide emissions from these fuels at relatively low cost compared with other methods. Sulfur emissions are potentially more serious than the soot problem in heavy oil use,
- f) emulsions can also reduce general boiler maintenance costs through cleaner combustion; and
- g) even without any efficiency increase by the use of emulsions of No. 6 oil, if the soot can be reduced significantly, a savings of as much as 23% in costs over the use of No. 2 oil could be possible. This would represent an annual saving to Goddard of \$300,000 a year. In order to meet environmental standards on No. 6 oil without emulsions, capital investments in stack scrubbing systems of over \$1,000,000 would be required.

In the next section we will discuss the modifications of experimental techniques that we used for Phase I that were necessary to perform tests for the heavy residual oils.

## II. Experimental Techniques and Instrumentation

The basic instrumentation for this study is described in our Phase I final report. In this section we will discuss the necessary modification to study emulsion with heavy residual oil (No. 6).

We now summarize the boiler instrumentation. A schematic of the boiler instrumentation is shown in Figure 1. The platinum resistance thermometers (A) measure the temperatures of feed water, output steam, and stack gases to within  $0.5^{\circ}\text{F}$ . Output steam is monitored by the differential pressure transducer and orifice plate (B) which, taking into account fluctuations, should be within 0.6%. Complete gas analysis, using the Hewlett Packard gas chromatograph (C) measure exhaust gas composition to within 0.1% accuracy. The amount of particulate emissions are determined by means of the EPA Train (D). Water and oil flow into the burner are measured by turbine meters (E) within .25%. The S-shaped pitot tube (G) determines the velocity in the stack with an accuracy of 7%.

In order to determine enthalpies of steam accurately, a pressure transducer (I) was installed in the steam line.

Two types of oil were tested. One was a low sulfur residual oil which was most like a blend of No. 2 and high sulfur No. 6. The second type of oil was a high sulfur No. 6. The properties of these oils are given in Table I. We now discuss the basic measuring techniques.

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### A) Thermal Efficiency

Our basic thermal efficiency study was conducted in exactly the same manner as our Phase 1 report. We made one major modification on the fuel flow meter. We found that the turbine meter was not acceptable to measure the No. 6 oils since the No. 6 oils carried suspended particulate matter which would damage the turbine blades. Too much filtering would make No. 6 oil more like No. 2. We decided, therefore, to go to a less-viscosity-dependent meter and chose a positive displacement meter identical to the ones installed in Goddard's fuel lines. During the latter part of the testing, we obtained a calibrated Niagara meter which was used to recalibrate all the Goddard meters. Though we did not get the one quarter percent accuracy in fuel readings that we expected with the turbine meter, we probably had about one half percent accuracy with these positive displacement meters and they performed satisfactorily.

In addition, during the latter part of the testing, after our data analysis had shown the problems with the pressure transducer in terms of electronic drift in the zero point, we decided to go to a different arrangement for measuring steam flow.

The new system employed two pressure transducers which were alternately monitored and while the first one was being monitored - measuring the differential pressure across the orifice plate - the second pressure transducer was being zeroed. Subsequently, the first pressure transducer was zeroed and the second pressure transducer was monitored.

This system of alternately measuring and zeroing of pressure transducers was continued throughout the data taking. This procedure proved to be very satisfactory in automatically correcting for zero drift of the pressure transducers.

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The Wang computer was modified for this phase also in that its memory capacity was doubled, this was required in order to monitor additional temperature and pressure transducers. The expansion of the memory obviously allowed expansion in the program which allowed us to take into account the two pressure transducers and the data monitoring.

The additional temperatures which were monitored were the wall temperature (in order to determine radiation and convection losses) and the oil and water temperatures (in order to obtain reliable heat input).

In order to determine the additional heat input due to heating the oil and water, we had to subsequently determine the heat capacity of the oil and thereby determine the heat input due to heating of the oil and water. This heat capacity of the oil was determined in the laboratory here at Adelphi using standard calorimetry techniques.

The BTU heat content of the oil was determined by bomb calorimetry techniques here at the laboratory at Adelphi University.

## B) Soot Measurement

In order to measure the particulate emission from the combustion of the fuel oil, the stack gases were passed across a series of heating filters. The speed at which these gases were pumped were adjusted to have isokinetic sampling so that the concentration that was measured was representative of that in the stack. For most of the tests, we used the standard size filters that were given to us by the Research Appliance Corporation.

A problem arose in determining soot in that only small amounts of soot were collected on these filters which were about 2 inches in diameter. The amount of soot collected was small, such that errors in weight determination could result in substantial errors in the soot analysis. For example, a 10 second delay in transferring the dried filter paper and soot from the oven to the scale could cause a substantial change in the weight of the soot due to water absorption.

When we went to much larger filter paper, as we did in the latter part of the tests, about 4 inches in diameter, we found that this effect was negligible and the soot measurements were reproducible which was not the case in some of our earlier runs

Soot data was determined in terms of milligrams per standard cubic foot dry which represents the basic concentrations in the dry stack gases.

#### C) Measurement of $\text{SO}_2$ Concentration in the Stack Gas

Two wet chemical methods (5,6) for determination of  $\text{SO}_2$  concentration were tried. Both methods involved trapping  $\text{SO}_2$  in an aqueous solution, followed by titration. Neither of these methods were found to be satisfactory because 1) it took a long time to collect samples, 2) titration was tedious and the end point was erratic and hard to determine, 3) the whole process took a long time and 4) the results were not dependable.

The method of choice was found to be a gas chromatographic determination<sup>(7)</sup>. Analysis using a column packed with phosphoric acid and acetone-washed Porapak QS gave satisfactory results. The column length was 18 inches, and the column temperature was 70°C. Originally gas samples had to be injected three different times on different columns to get complete analysis of the stack gas composition. However, with the use of the column changer, it was



possible to obtain complete analysis of the stack gas composition with one injection. This method was simple, fast (only 10 to 15 minutes instead of hours) and gave consistent results.

Particulate matter emitted when we added the soda ash to the fuel, mainly sulfate, was collected on the stack sampler filters and weighed. The sulfur removal measurements are considered to be reliable throughout the series of the tests.

In addition, to determine deposition of the tubes, a series of stainless steel bands were clamped around several sample boiler tubes such that the various aerodynamic flow patterns in the furnace could be sampled in terms of their deposition. Subsequently we will discuss the results of this sampling procedure. It was deemed to be a satisfactory method of determining boiler depositions, not only for the soda ash process to remove the sulfur dioxide, but also for deposition of the soot for the heavy oils.

### III. Discussion of Results

#### A) Thermal Efficiency - Low Sulfur Oil

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The low sulfur oil that was used was a blend of No. 2 and No. 6 oil. Its characteristics were more like No. 4 than No. 6. An analysis is given in Table I.

In our initial tests with this oil, we obtained several results with abnormally high thermal efficiencies (88-89%) for emulsions. During a two day sequence, the emulsifier was turned off and the efficiency appeared to drop (83-84%). Though this appeared to be significant evidence of improved thermal efficiency with emulsions at the time, we could not rule out instrument error.

Subsequent analysis of this data did not turn up any errors, but also did not rule out any. Since this data involved measuring output BTU's/input BTU's, these errors would have to be in steam flow and/or fuel flow. We did have evidence that there was electronic drift in the pressure transducers. Although it had apparently not been carefully monitored during these tests, the electronic drift would have had to be exceptionally large to account for the surprisingly large efficiency values. Subsequently the transducers were replaced. In our final data analysis, we examined efficiency in terms of heat-losses through the following equation

$$\eta \text{ (thermal efficiency \%)} = 100 - \text{total heat losses}$$

A check on our measurements was the heat balance, efficiency (output/input) + heat losses = 100. In many runs we did not obtain a complete heat balance. This was due, most likely, to the fact that certain losses which were known to exist such as possible leaks in the steam pipes and others (see discussion in Appendix B) could not be taken into account. A typical heat loss calculation is shown in Appendix B.

Figures 4,5,6 are a compilation of all our efficiency data, including output/input, heat loss, and heat balance data. We see that in many of the No. 4 oil runs with high efficiency, the total heat balance is greater than 100%. In principle, this should not happen. Random errors are expected to be no more than 1%. In earlier runs we did note that electronic drift of the pressure transducers could lead to errors of 2%. Once they were replaced by a new transducer system, the error was probably no more than 1%. Therefore, we are led to conclude that those high efficiency runs which lead to a mismatch heat balance of 106, 107% must be in error.

This error could have arisen, as indicated, from electronic drift in the pressure transducer. Also, during this period, the Keithley voltmeter became inoperative; it may have, without our being aware of the drift, either caused or contributed to the electronic drift of the pressure transducer. Subsequently, the Keithley was repaired and the pressure transducers replaced so that subsequent runs were o k. Therefore, in Figure 7 we have plotted efficiency as a function of excess air for oil and emulsion for low sulfur oil using the stack loss data. We see that in this data emulsions and oil yield about the same thermal efficiency.

#### B) Thermal Efficiency for No. 6 - High Sulfur Oil

This data is plotted in Figures 8,9. Stack loss and output/input methods are consistent so that these curves are more reliable than the low sulfur data. Again, the result is that the difference between emulsions and oil is within experimental error (See Figures 10,11).

### C) Effect of Emulsions on Particulate Emissions

#### (1) Soot Emissions from Low Sulfur Oils

Figure 12 shows the effect of using emulsions for the low sulfur oil. Data is plotted in terms of soot measured in milligrams per cubic foot and percent excess air. As can be seen there is approximately a 10% reduction in soot when using the emulsions. This could be a little more than 10% which we will discuss later. As the excess air is increased, the soot produced in the emulsions becomes approximately the same as that from the oil

From this, it is clear that though emulsions will reduce soot for the Goddard steam atomizing system, for the low sulfur oil, reductions are not as substantial as has been observed at other systems which employ air atomizing or pressure atomizing.

We see that we can reduce the excess air for emulsions if we reduce the excess air to run at the same level of soot for oil as emulsions. That is, reduce the excess air from approximately 38% to 28% from our efficiency curves this will amount of about a 1% increase in thermal efficiency. This is about the best that can be hoped for.

Conceivably, this could be improved by upping the pre-heat temperature on the oil when an emulsion is run. Later on we will discuss some evidence for a significant effect of pre-heat temperatures and soot emissions.

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## (2) Soot Emissions from High Sulfur Oil

For the high sulfur oil, data is shown in Figure 13. Again we have the same type of curve that we had for the low sulfur oil. For low excess air we have about a 13% reduction in soot. For high excess air the emulsion point crosses over the oil and its effect is slightly higher than the soot levels for the oil.

We note that when we decreased our pre-heat temperature on the oil for an emulsion run from 90°C to 75°C, soot increased by about 30%. That is over 40% excess air our soot increased from 3.5 mg per cubic foot to over 4.3 mg per cubic foot. We also know that the viscosity of emulsions is higher than that of oil. Also the surface tension is temperature dependent. These two parameters are very important in atomization which affects soot production.

It is not inconceivable, therefore, that part of the reason that we do not see substantial soot reductions for the high sulfur oils was due to the fact that we did not operate at pre-heat temperatures sufficiently high. One can expect that if we could push the pre-heat temperature of the oil above 90°C when an emulsion is run, we might be able to get soot reductions of the order of 30%. From a practical standpoint, however, with Goddard's systems, they are not too well set up to run at such high pre-heat due to carbonization and dirtying of burner tips. However, this is something that must be taken into account. Again, if we reduce our excess air for oil to operate at the same levels as emulsions, we can foresee about a 1% increase in thermal efficiency.

The standards for particulate emissions at Goddard according to Mr. Harold Die of Maryland Environmental Protection Agency, is .030 grains per standard cubic foot (corresponds to 1.94 mg/SCFD). In addition, there can be no visible smoke. It is not possible even with careful tuning, to run within the standards for particulate emissions for both oil and emulsion with low and high sulfur oils. The advantage of emulsions, therefore, is simply the fact that we may run at slightly lower excess air without visible smoke, which may translate to about a percent increase in thermal efficiency.

We now want to talk about a very important point which was discovered at the end of our data analysis, which might explain the relatively low effect that emulsions had on soot reduction. Figure 15 shows a curve reproduced from the report by Bob Hall of the EPA showing for an air atomizing system the effect of emulsions on soot reductions. We note that as he increased the water concentration soot levels go down. We also refer to a paper by a Norwegian author which again shows very substantial reductions in soot for emulsions with a pressure atomizing system.<sup>(9)</sup>

The question then arises why are we not seeing this for the steam atomizing system at Goddard. It occurred to us that there might be some possibility that the design of the steam atomizer at Goddard is such as to produce an emulsion already before the addition of the pre-emulsifier. In fact, we subsequently took some photomicrographs of a steam/oil spray from our laboratory steam atomizing system. As is shown in Figure 14 it does indeed produce an emulsion with about a 20% water concentration, an emulsion which is almost as good as the ultrasonic unit.

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In the burner that is used at Goddard the steam and oil are mixed in a cavity before the total fluid is sprayed into the boiler. This system will very likely produce an emulsion very similar to what we saw in the laboratory. It would seem, therefore, that the steam atomizing system itself is making an emulsion and, therefore, by adding more water, we are somewhat improving the atomization properties as is evidenced by the improvement in soot reduction for increasing water (as in Hall's report) but we are not making a dramatic improvement because we are dealing with an emulsified fuel already. Also particulates are only 0.2 - 0.4% of the oil. It is possible that a large portion of this is non-combustible ash.

#### D) Sulfur Removal

In the course of Adelphi's experiments on the use of a coal/water/oil emulsion as a liquid fuel, it was discovered that it is possible to remove the sulfur dioxide from the combustion gases generated by this fuel (which is equivalent to a heavy residual oil), by the addition of soda ash. Since soda ash is soluble, it can be first dissolved in the water, which is then added to the oil to form an emulsion. Because of this high solubility in water, soda ash appears preferable to limestone as an additive. (Our own preliminary tests with limestone reveal that it is a feasible, if less desirable alternative)

When the water in droplet form is vaporized in the furnace, and the fuel droplets blown apart, tiny crystals of soda ash are formed which follow the flow patterns of the sulfur dioxide gases and absorb them very efficiently, leaving a harmless residue of sodium sulfate fly ash, which can be collected by precipitators in the stack. Figure 16 shows the results of sulfur dioxide removal with this process both in the laboratory

and during an 8 hour test with Goddard boilers. As can be seen, the efficiency removal is very high, approaching 100%, a result very difficult to obtain by such other means as limestone injection or fluidized bed combustion. The direct application to heavy residual fuels with a high sulfur content is apparent. The sodium sulfate ash formed was a dry powder easily scraped from the boiler tubes. The fly ash was approximately 1 $\mu$ m particle size and could be precipitated out. Additionally, the hydrated sodium sulfate product can be used for chemical storage of solar energy.

Figure 17 indicates that there might be a slight decrease in boiler efficiency after a time due to tube deposits.

Appendix C shows the details of the sulfur removal calculation.



## E) Boiler Tube Deposits

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## (1) Soot

During our testing period, we periodically examined the interior of the boiler for deposits. Photographs of the interior were compared with those before the start of the tests when the boiler was cleaned. The deposits were examined after a running time of from 4-5 months. Figure 18 shows a chronological sequence in the combustion history of the boiler tubes. Figure 18A is a layout of our test area. Though there was a slight indication of a cleaner burn with emulsions (not shown) deposits from No. 6 oil could be scraped off but with some difficulty. Our main interest was to compare deposits with and without soda ash. Since soot reduced only about 13%, the clean-up effect is not dramatic on the tubes and the effect on heat transfer is negligible. There was a cleaning of the burner tips, however, when emulsions were burned.

## (2) Deposits from Soda Ash Process

After our soda ash run (8 hours duration) a sequence of photographs of the boiler interior were taken. Figure 19A shows a tube wall. Though the deposits were noticeable, the material was a white powdery substance which easily flaked off the tubes. There were no deposits on the floor indicating the particulate matter was small and entrained by furnace gases. Examination of sample tube 3 shows the nature of the deposits (Figure 19C). We note the comparison with soot deposition (Figure 19D). Figure 19E shows the tube scraped clean of sodium sulfate. The cleaning process was accomplished easily with a wire brush, much easier than cleaning the tubes of soot. In fact, when the soot was mixed with sodium sulfate particles it did not adhere strongly to the boiler tubes and they could be cleaned much more

thoroughly than when covered with straight oil emulsions. Figure 19F is a view of the stack during the soda ash run. A white cloud was emitted which could probably be eliminated by stack precipitators. The level of particulate emissions from the soda ash process was 70 mg per standard cubic foot. We would need to remove 97% of this to be within the Maryland EPA standards.

#### IV. Conclusions and Recommendations

It is clear, despite much of the literature, that for a Goddard type system of steam atomizing, an emulsion is very likely being produced prior to additional water being introduced, and that though we can gain some improvement, maybe a percent in thermal efficiency, and perhaps from 13 to at best 30% reduction in soot with increased pre-heat, it is not as dramatic an effect as one would expect from the literature. Therefore, in terms of soot reduction and thermal efficiency, with Goddard's system, a pre-emulsifying system is not warranted. There is another side to the coin, however, in terms of sulfur removal. Here the results were very encouraging. We obtained 100% efficiency of removal and we did not obtain a severe deposition on the boiler tubes. If a fairly inexpensive precipitator could be found then it might be economical to run the very high sulfur oil which is substantially cheaper than the low sulfur oil using the pre-emulsifying system. It would not be practical to dissolve soda ash and send it in with the steam atomizing system. So the pre-emulsifying system has substantial benefits in terms of sulfur removal. At the current price of soda ash (\$60 per ton) we would expect to add about 1-2¢ per gallon to the cost of the fuel.

Another point is that for those systems in NASA which use air or pressure atomizing and, for example, are hot water generators, it would not be practical or economical to put in a steam atomizing system. Here, it would be economical to use an emulsifying system. In our preliminary survey of emulsifying systems, we see that Tymponic system which was used for the NASA tests, though readily available and on the market, is far too expensive for the benefits to be obtained. Fortunately, there are several other companies in the market which produce satisfactory emulsifying systems at a much lower cost. For example, Lightin' Mixers, Inc.

produces a system which would cost somewhere from \$3000 to \$5000 and our laboratory studies have shown that this produces a very good emulsion. Additionally, there are colloid mills for the order of \$5000 which also produce a good emulsion, and there is a Japanese mixer, by Funken Co., which we have also been testing which would cost more than the colloid mill, but would allow the addition of pulverized coal to the oil, which could be of very substantial benefit when oil supplies become limited, as we know they will. Figures 20, show pictures of the Funken & Lightin' Mixers.

We, therefore, would recommend that a survey be done on NASA's installations to catalogue the number of air and pressure atomizing boilers which might be converted to No. 6 oil or already use it and estimate the overall cost of converting these to a relatively inexpensive emulsifier such as the Lightin' Mixer. If this looks economical, projecting our estimates of soot reduction, then it might be advisable for NASA to seriously consider this alternative.

Also, a study should be done on the cost of precipitators and the use of the soda ash sulfur removal process.

The present Maryland EPA standard for sulfur in residual oil is 1%. None of the oils tested including "low sulfur" oil met this standard. For the low sulfur oil we would have to remove about 15% of the sulfur to meet with EPA standards. This would require only 30% of the soda ash that was used in our high sulfur oil test. Particulate emission in this case would be around 20 mg per standard cubic foot which is not acceptable for No. 6 oil and a dust collector would be required.

Since we only need 30% of what is used for high sulfur oil the cost would be much less than a penny a gallon.

It appears, then, that this might be a viable process for the low sulfur oil. For the high sulfur oil, as mentioned earlier, we need large quantities of soda ash and some method of precipitating particulates would have to be found. This requires off the shelf equipment and the cost advantages of using No. 6 oil would result in short term pay-back.

One point that must be considered in the future is the possibility that the No. 2 distillate oil will not be available and that we must find a method of going to high sulfur oils. Compounding this difficulty is the fact that the Maryland standards as of July 1, 1980 will be 0.5% sulfur with the heavier oils. If this occurs, then we must go to a sulfur removal system to burn these heavier oils.

## REFERENCES

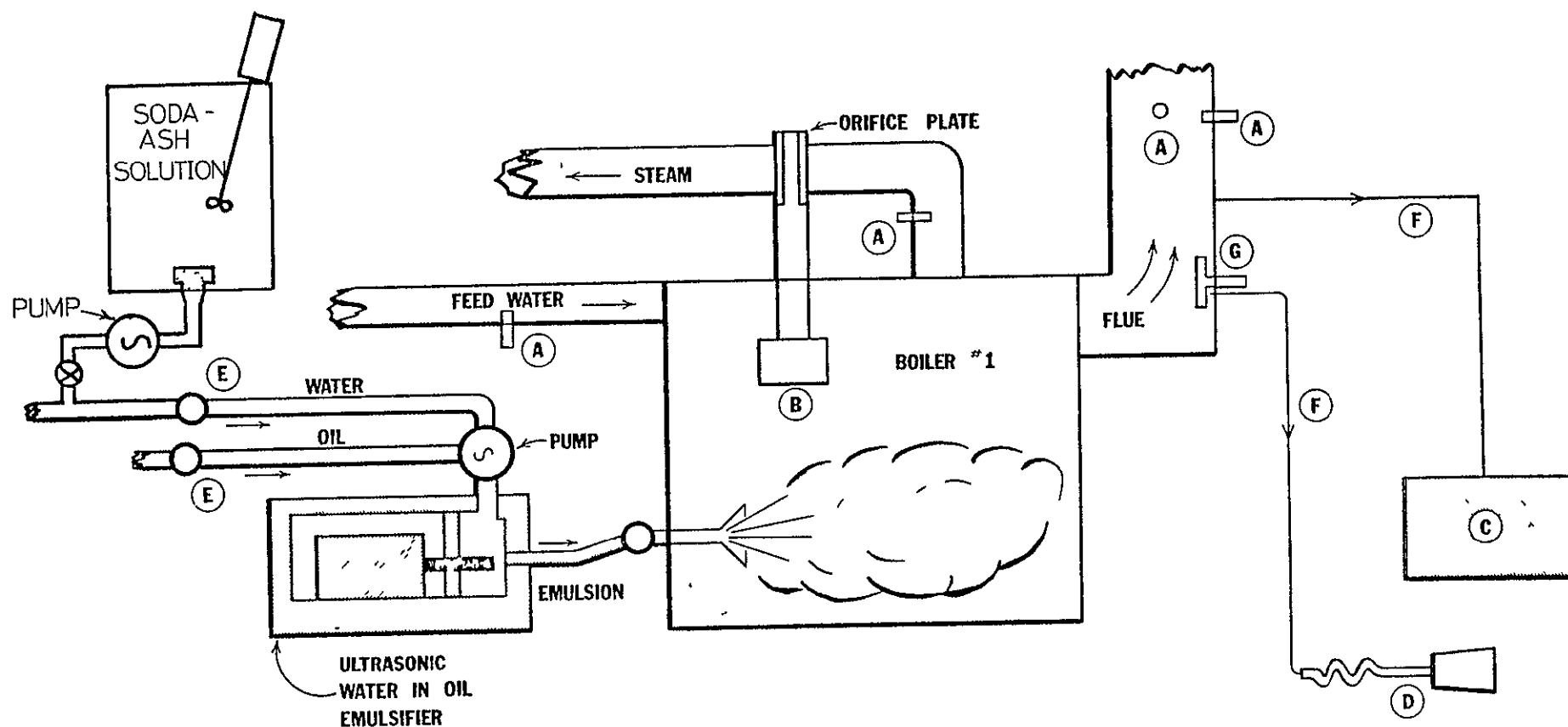
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- (7) "Method 8. Determination of SO<sub>2</sub>; SO<sub>3</sub> and sulfuric acid mist," Federal Register, vol. 36, no. 247; Research Appliance Company Train Staksamplr instructions Catalog No. 2343
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Table 1

CHARACTERISTICS OF TEST OILS

OIL SAMPLE	% CARBON	% HYDROGEN	% SULFUR	VISCOSITY AT 60°C (cp)	SPECIFIC GRAVITY AT 20°C	BTU/GAL 20°C
16-41	83.71	10.67	2.43	256	.978	149,415
16-11	85.22	11.62	1.54	30	.917	145,750
15-55	86.26	12.62	1.12			



**FIG 1**  
**ADELPHI - GODDARD**

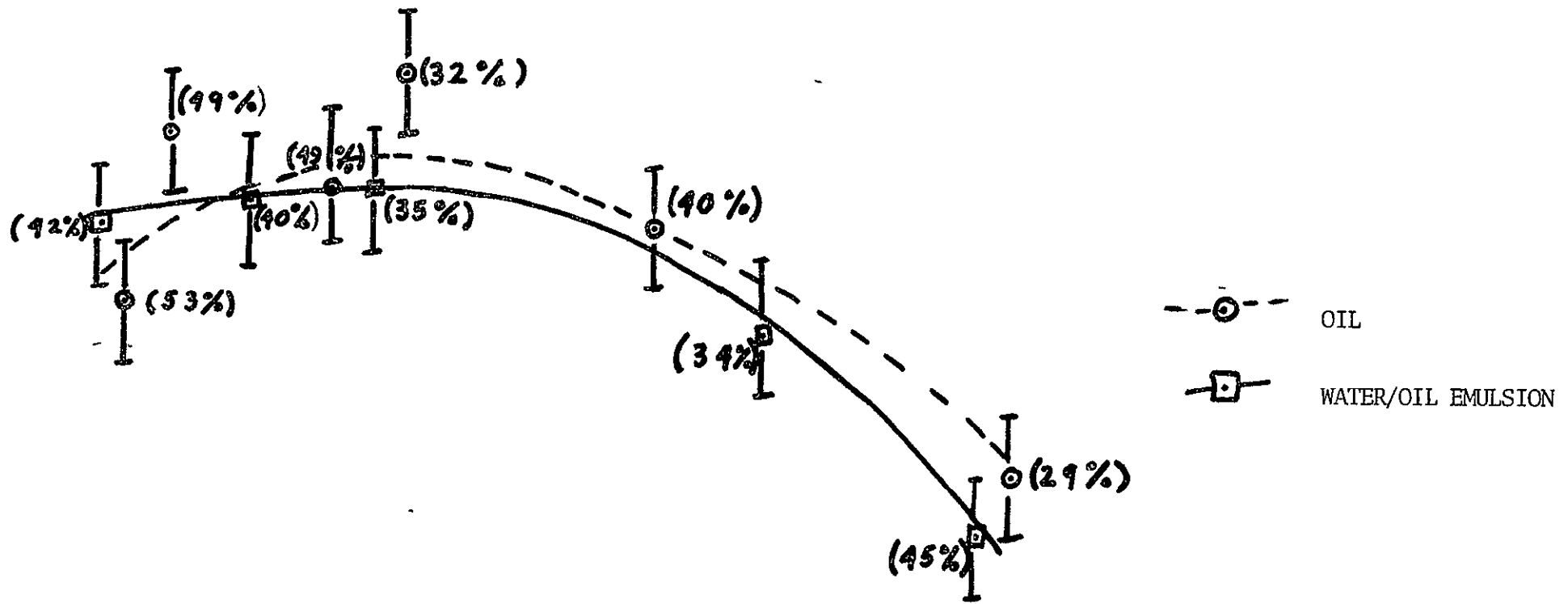
## EMULSIFIED FUEL PROJECT

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THERMAL EFFICIENCY WITH OIL AND WATER/OIL EMULSIONS  
INCLUDING % EXCESS AIR

Fig 2



# SMOKE NUMBER FOR OIL

Fig 3A

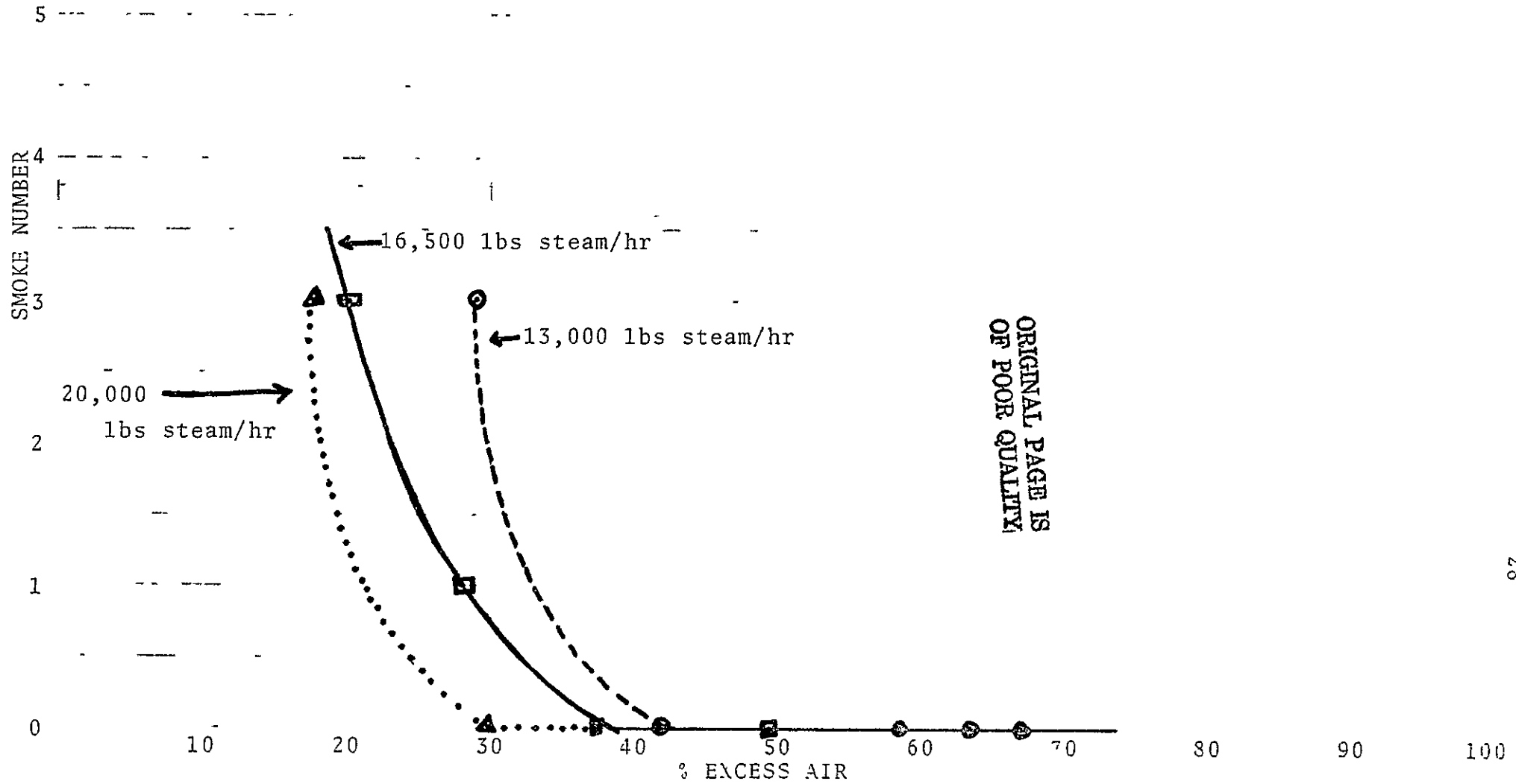


Fig 3B

SMOKE NUMBER FOR EMULSION

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Figure 4  
Thermal Efficiency (Input/out) and Losses

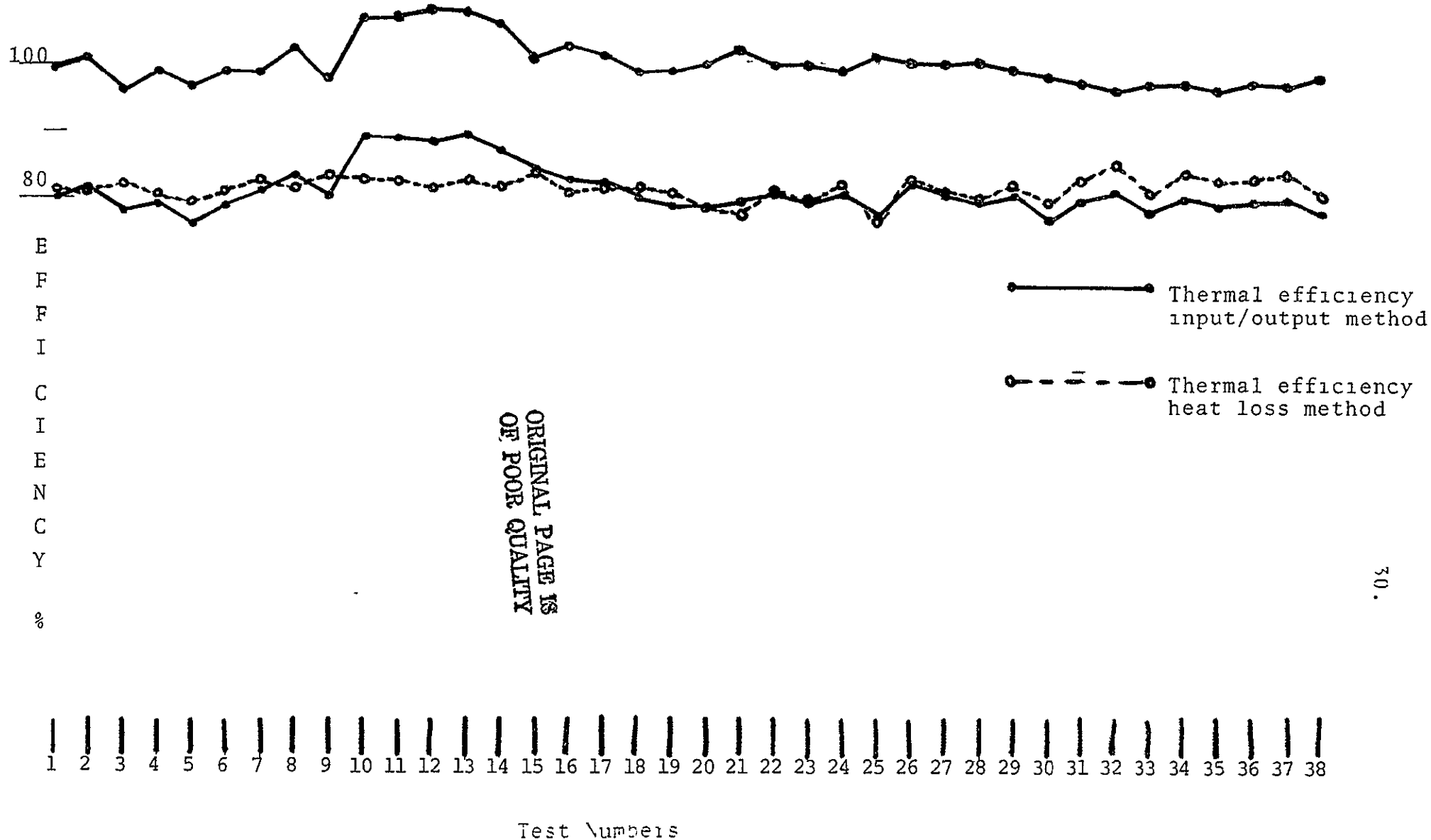
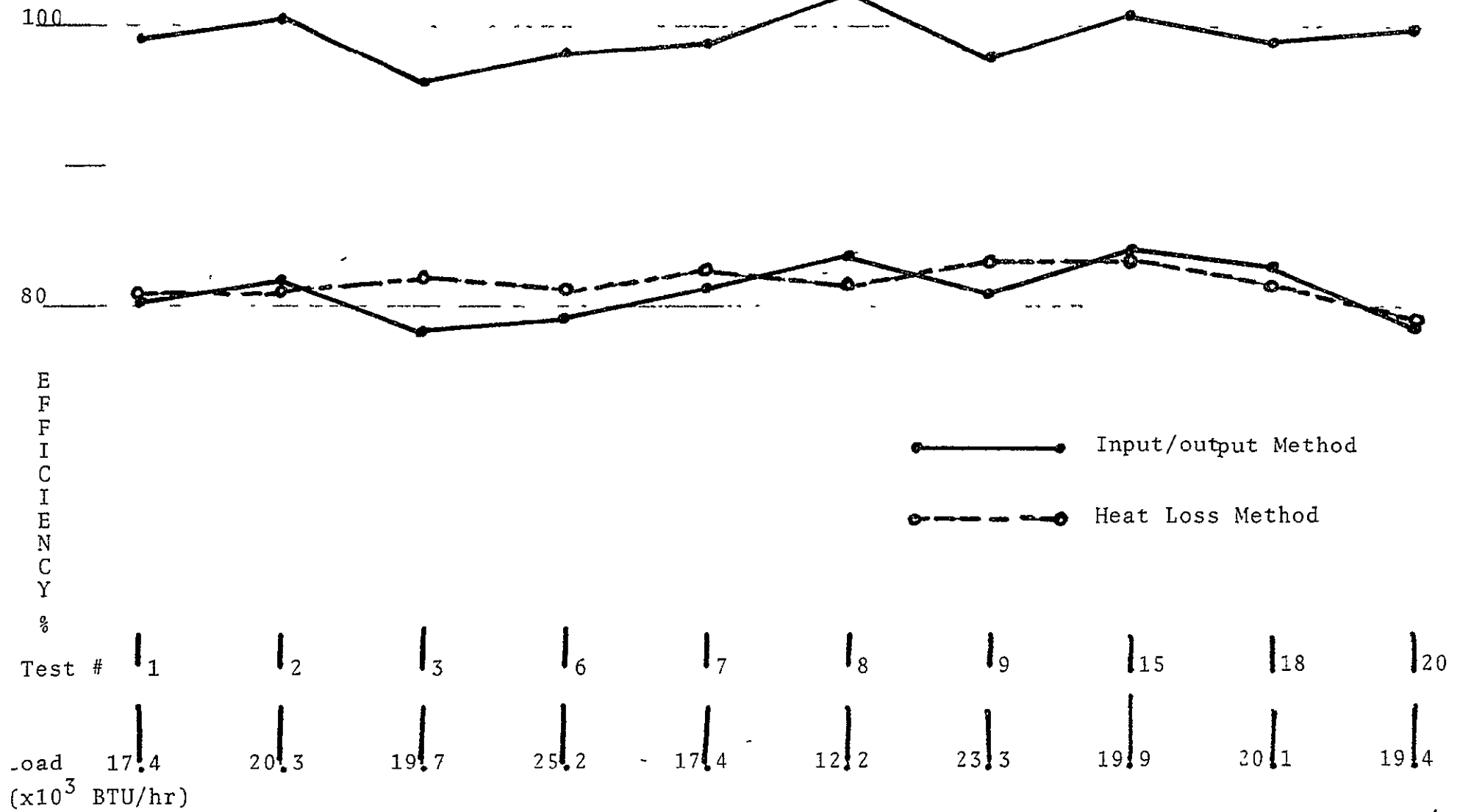


Figure 5  
No. 4 Oil  
Thermal Efficiency (input/output)  
& Losses



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Figure 6

No. 4 Emulsion  
Thermal Efficiency (input/output)  
& Losses

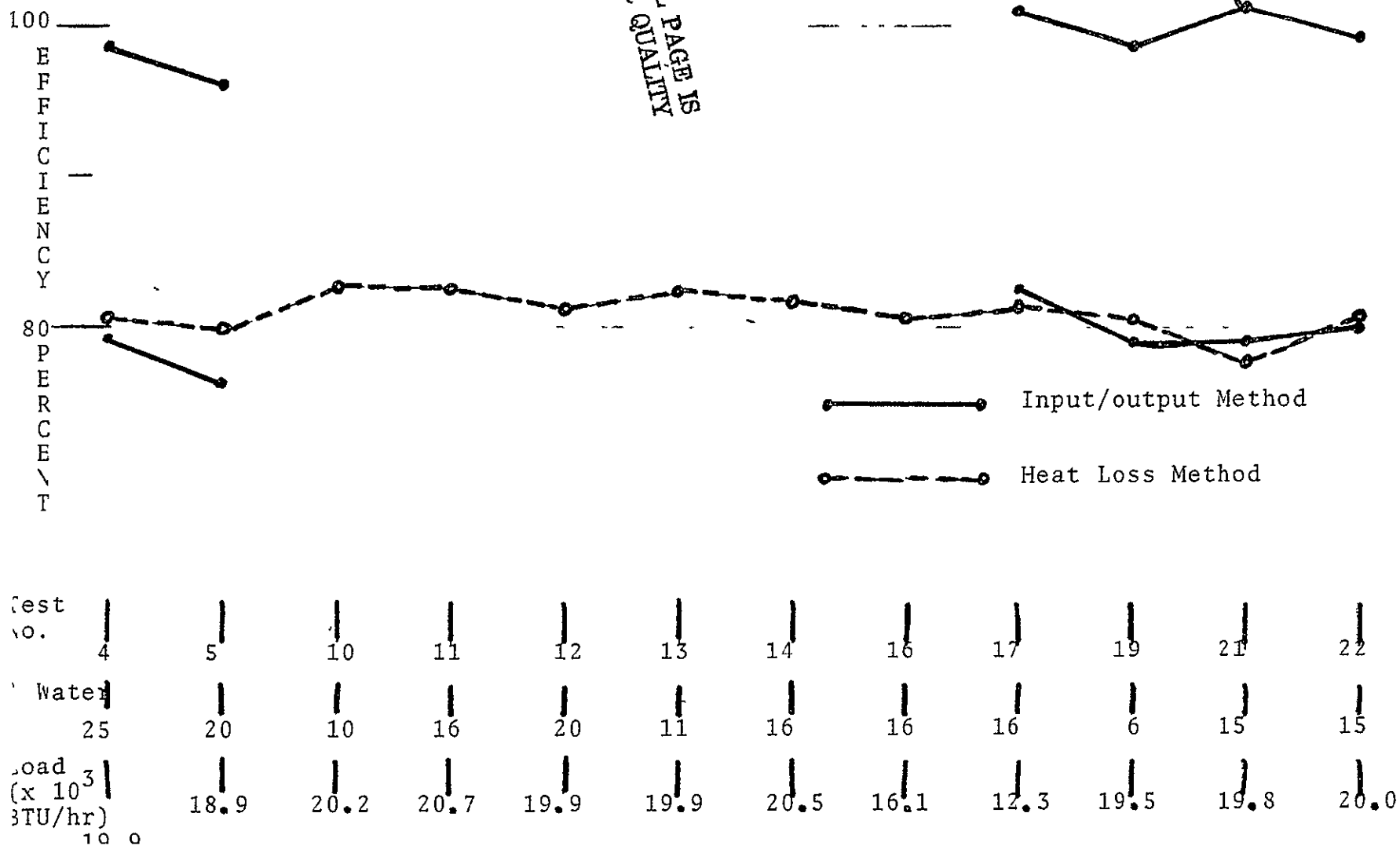


Figure 7

No. 4 Oil

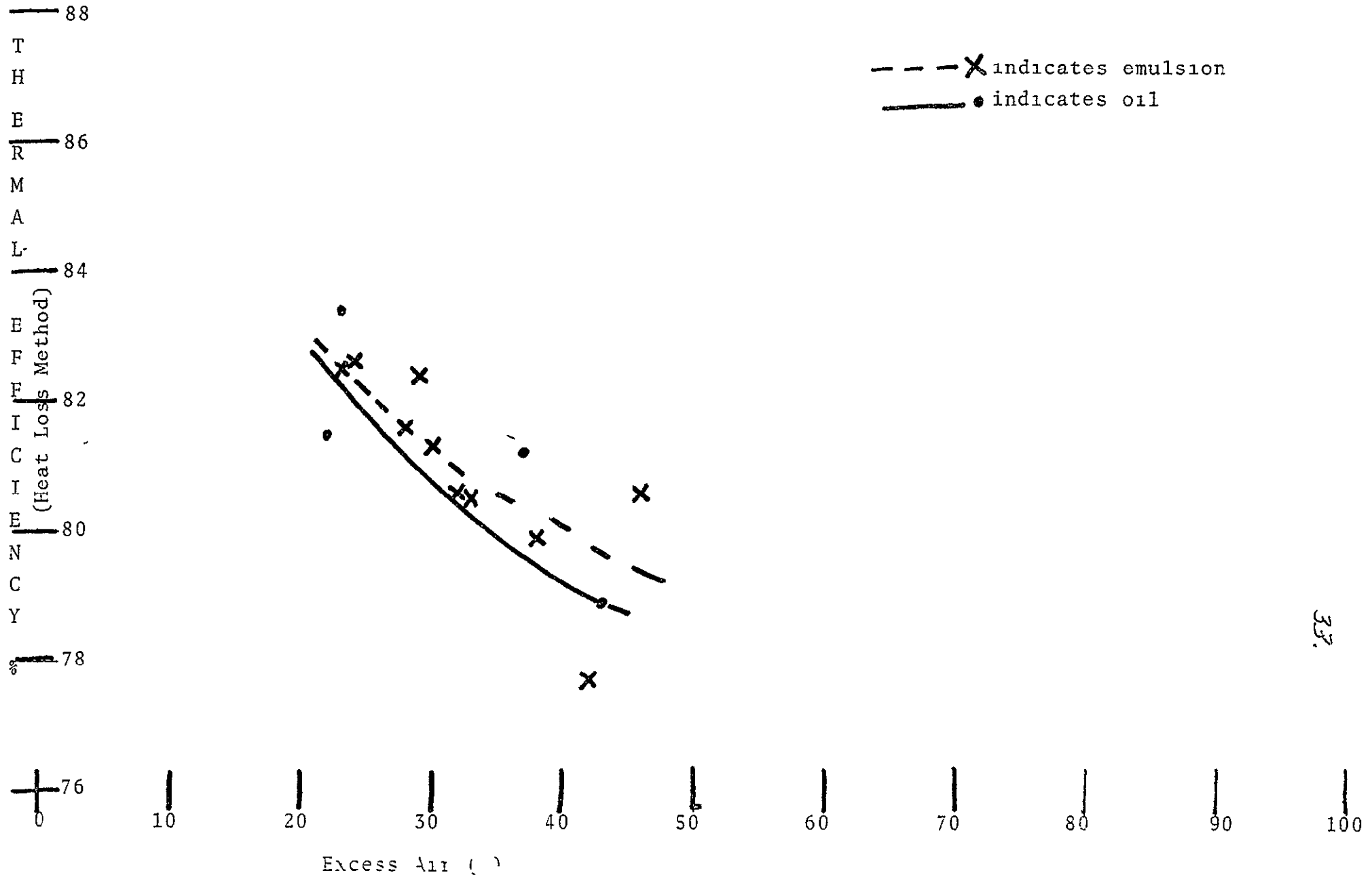


Figure 8

No. 6 Oil

Thermal Efficiency (input/output)  
& Losses

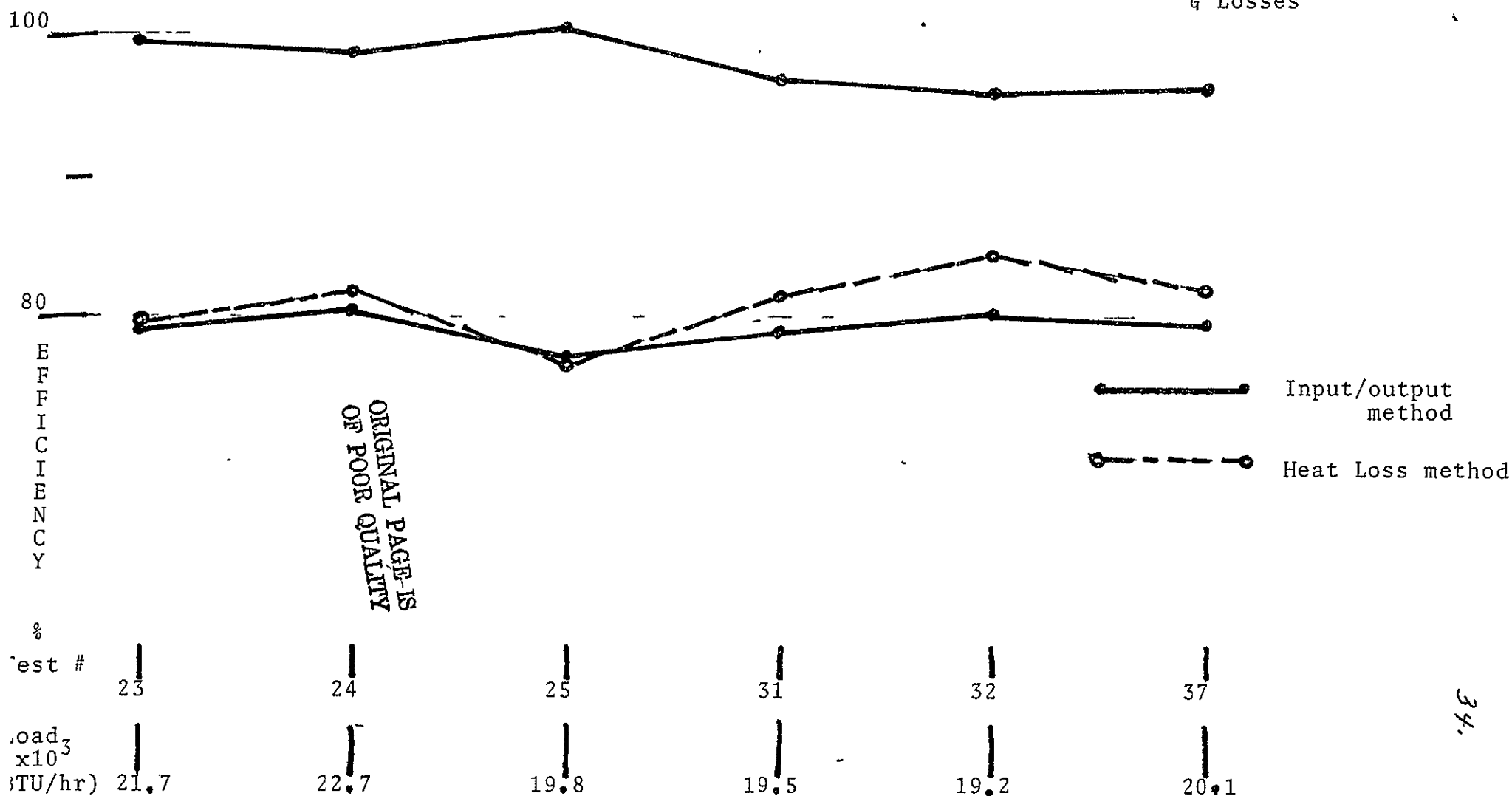




Figure 9

No. 6 Emulsion

Thermal Efficiency (input/output) &amp; Loss

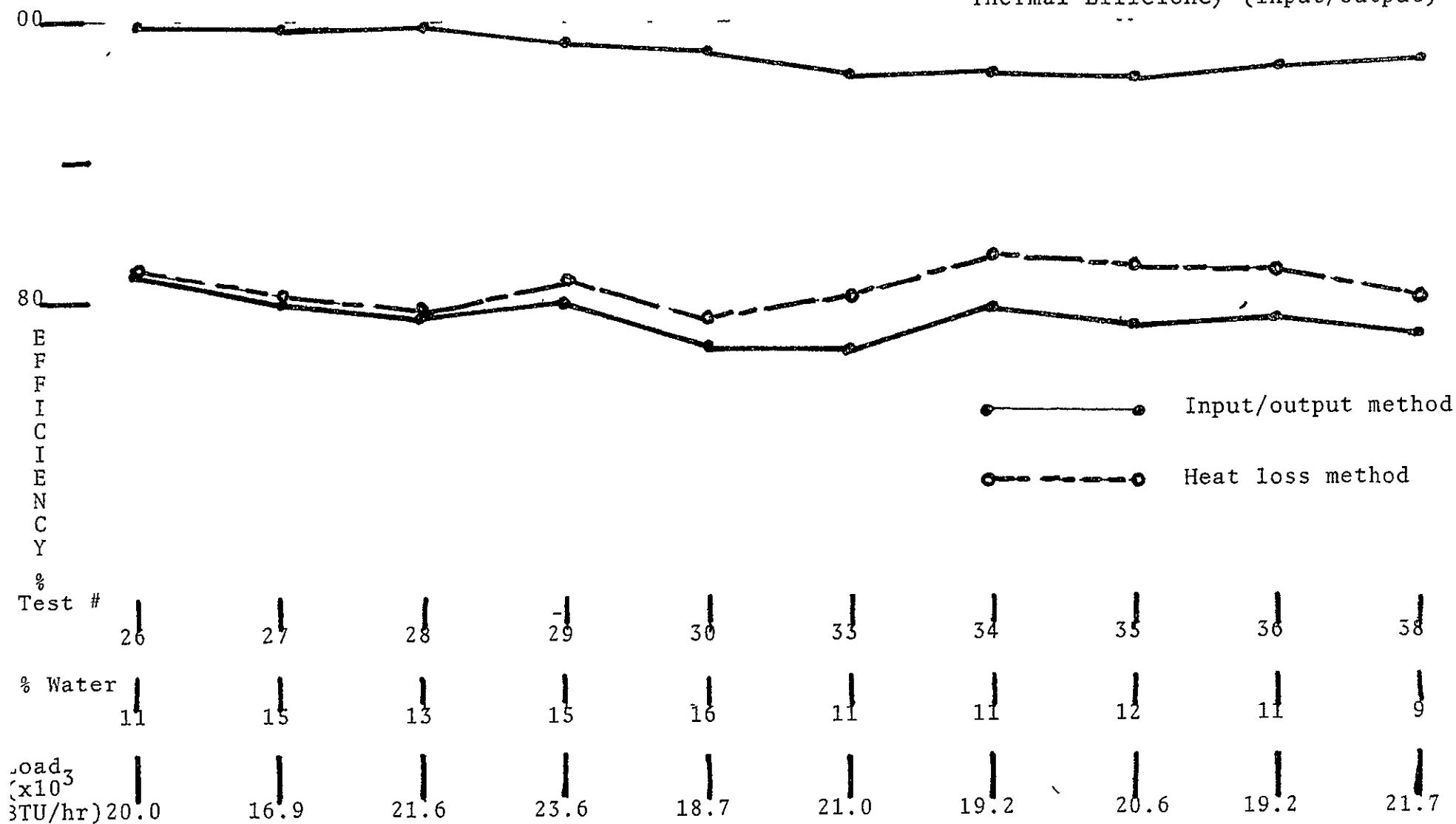


Figure 10

No. 6 Oil

— x indicates emulsion  
— • indicates oil

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C  
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Y  
%

(Input/output Method)

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EXCESS AIR (%)

36.

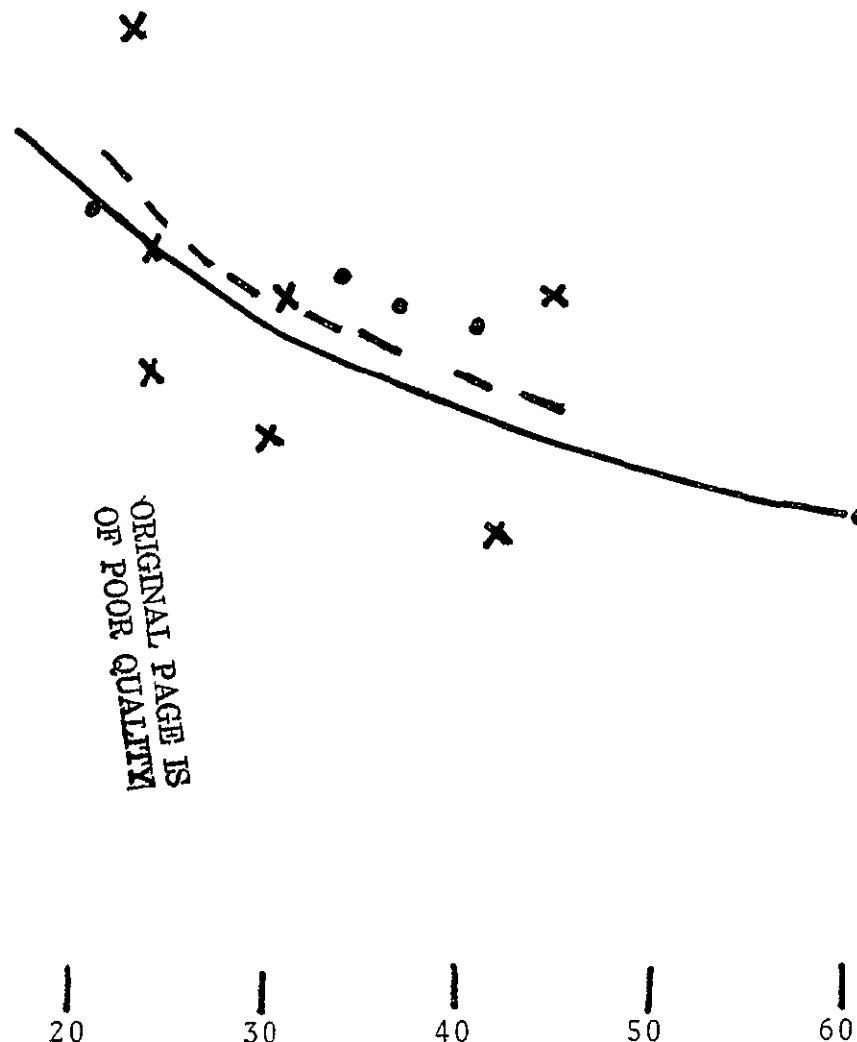


Figure 11

No. 6 Oil

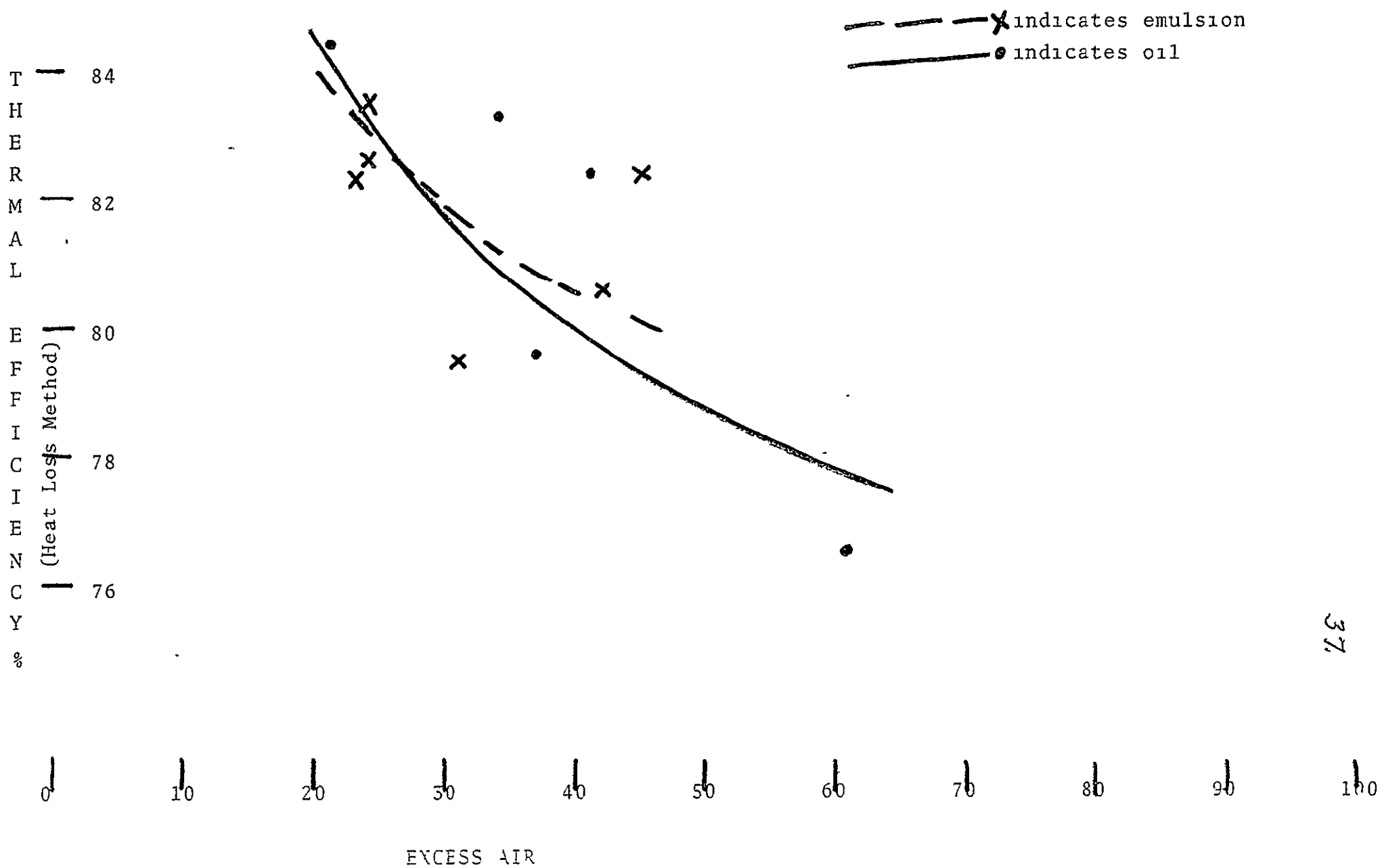


Fig. 12

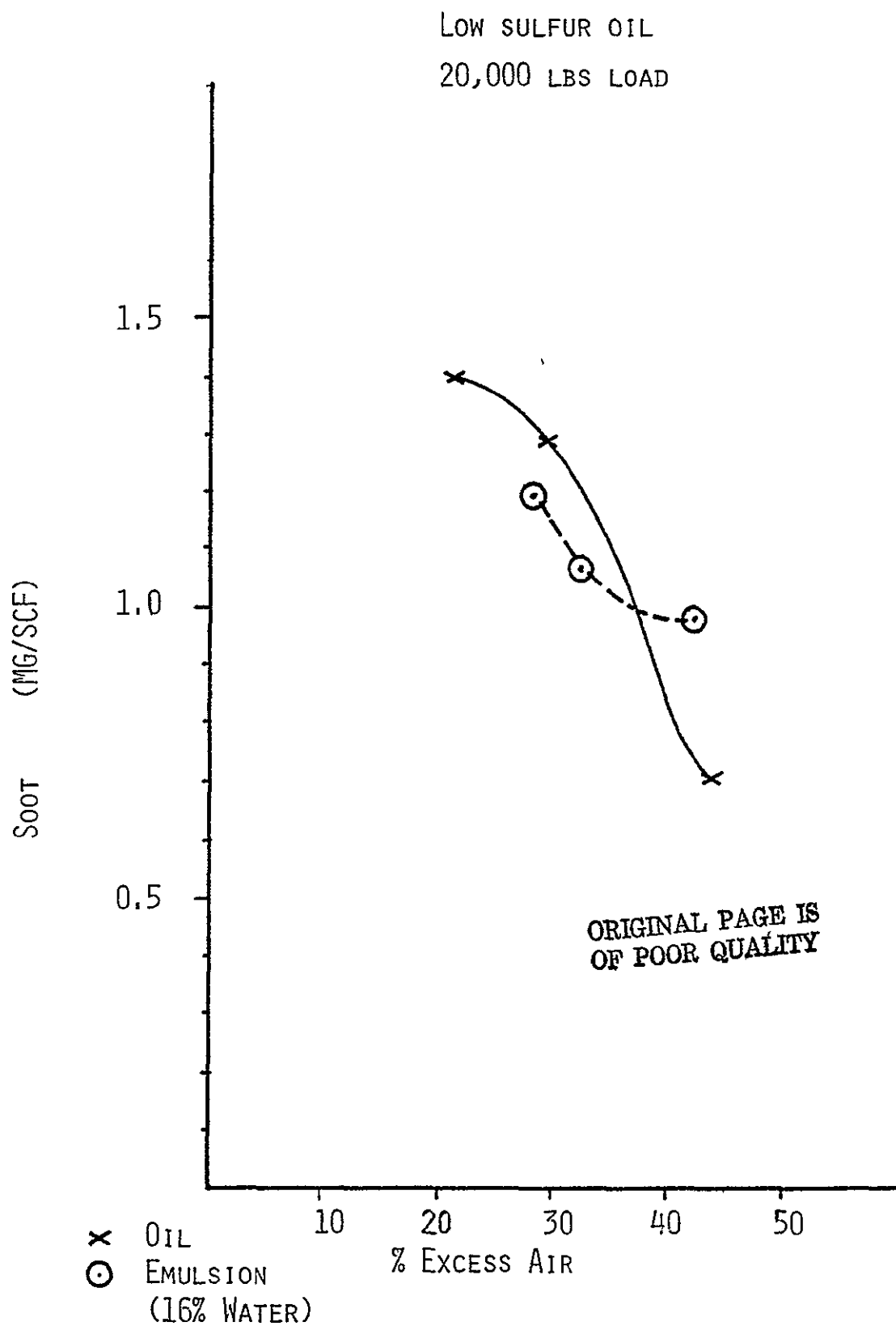
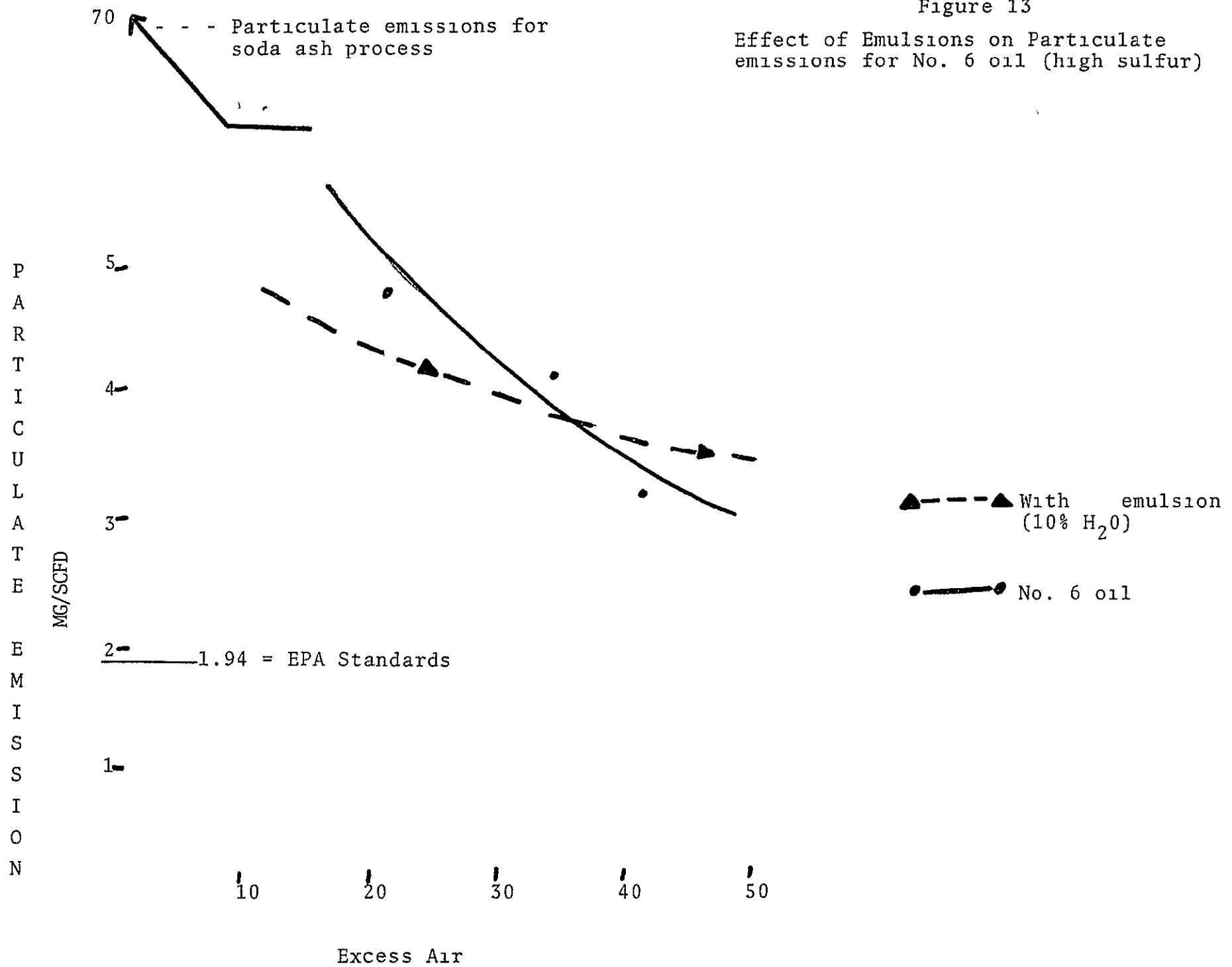


Figure 13

Effect of Emulsions on Particulate emissions for No. 6 oil (high sulfur)



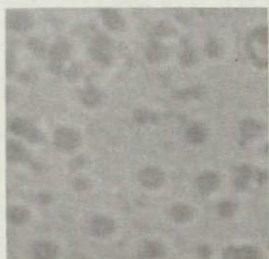


Fig. 14A  
Ultrasonic

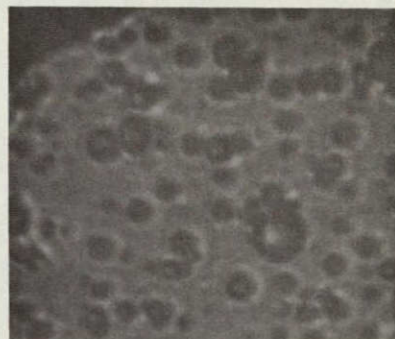


Fig. 14B  
Funken Mixer



Fig. 14C  
Steam Atomizer



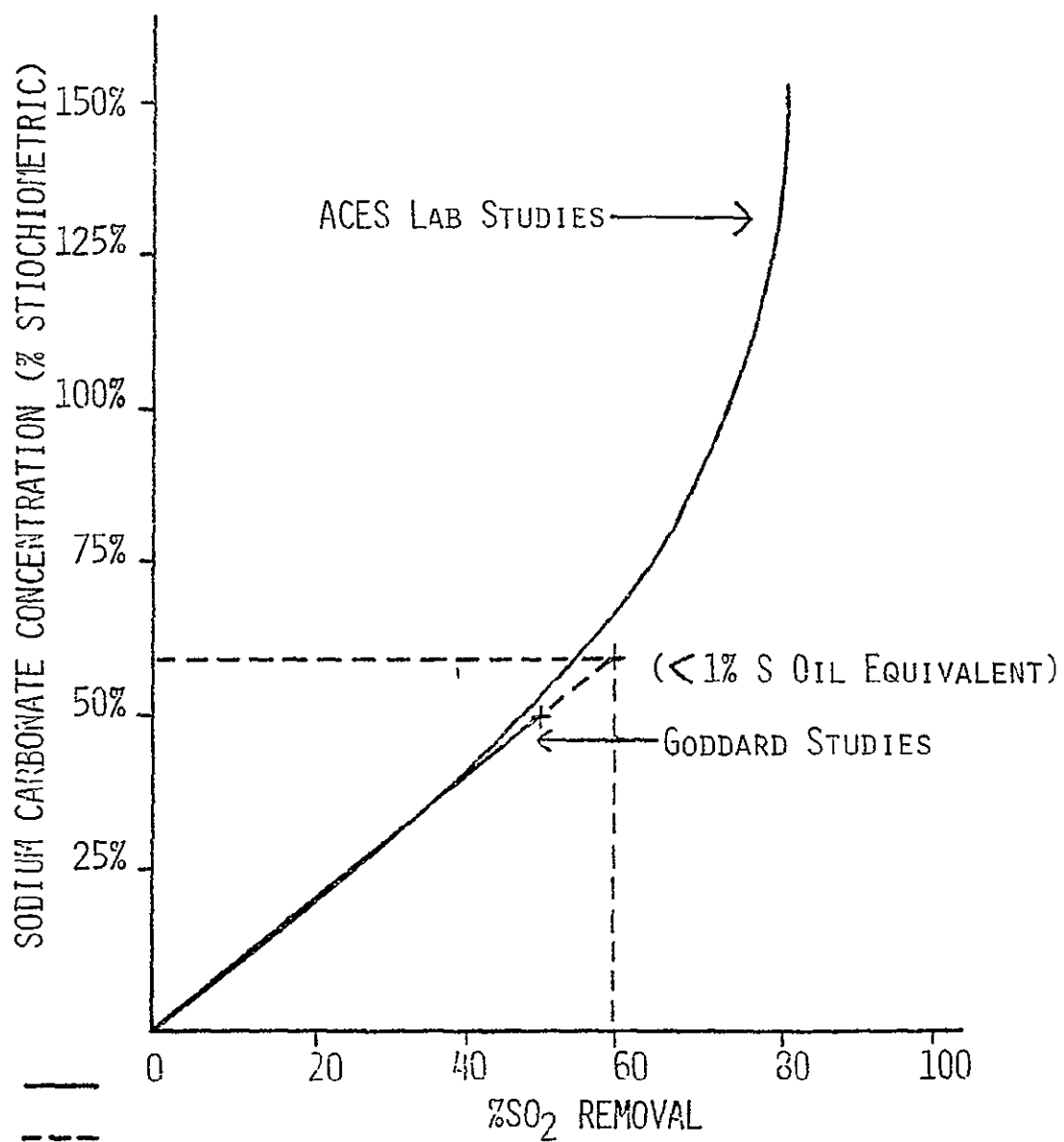
Fig. 14D  
Lightin' Mixer



Fig. 14E  
Pure Oil

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Fig. 16

EFFECT OF SODIUM CARBONATE ON  $\text{SO}_2$  REMOVAL

STACK TEMPERATURES FOR 8 HOURS  
OF EMULSION WITH SODA  
ASH ADDITION

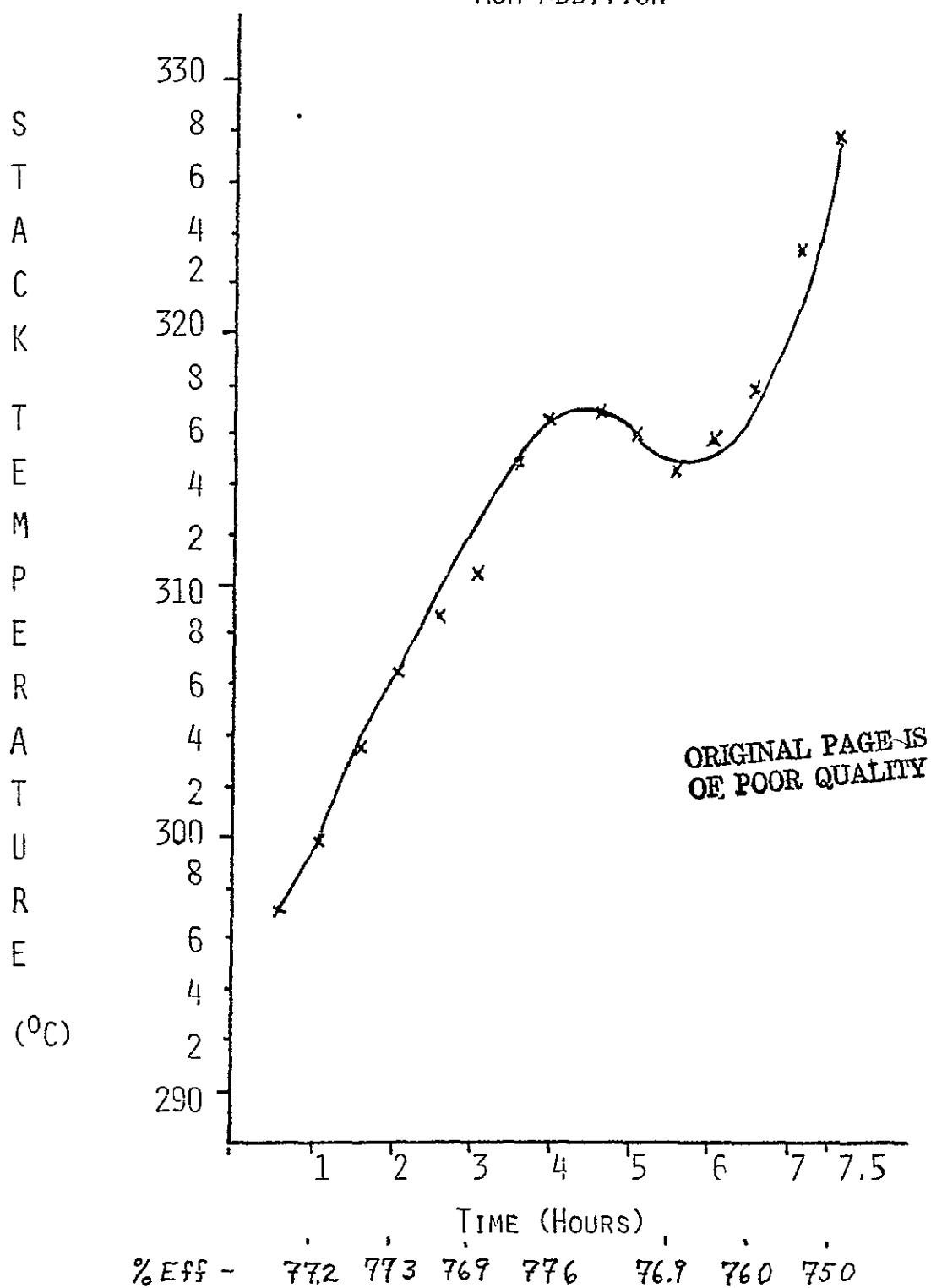
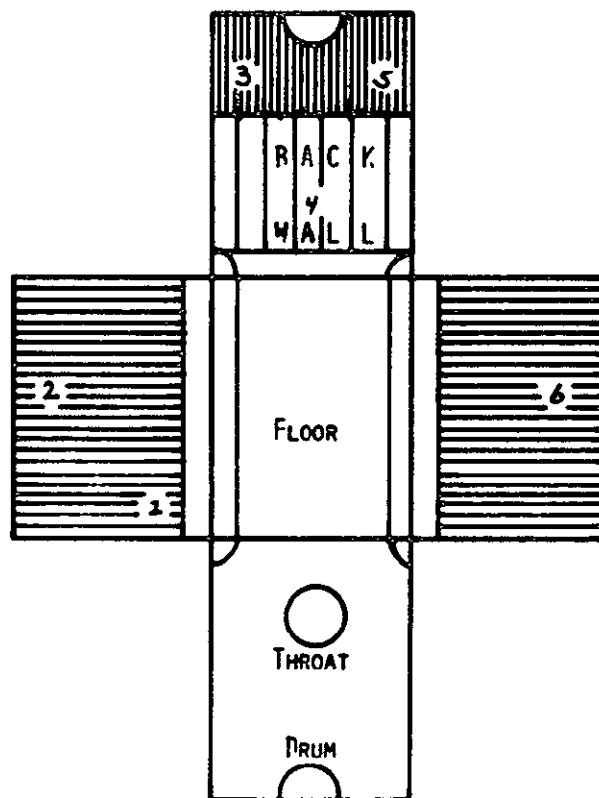




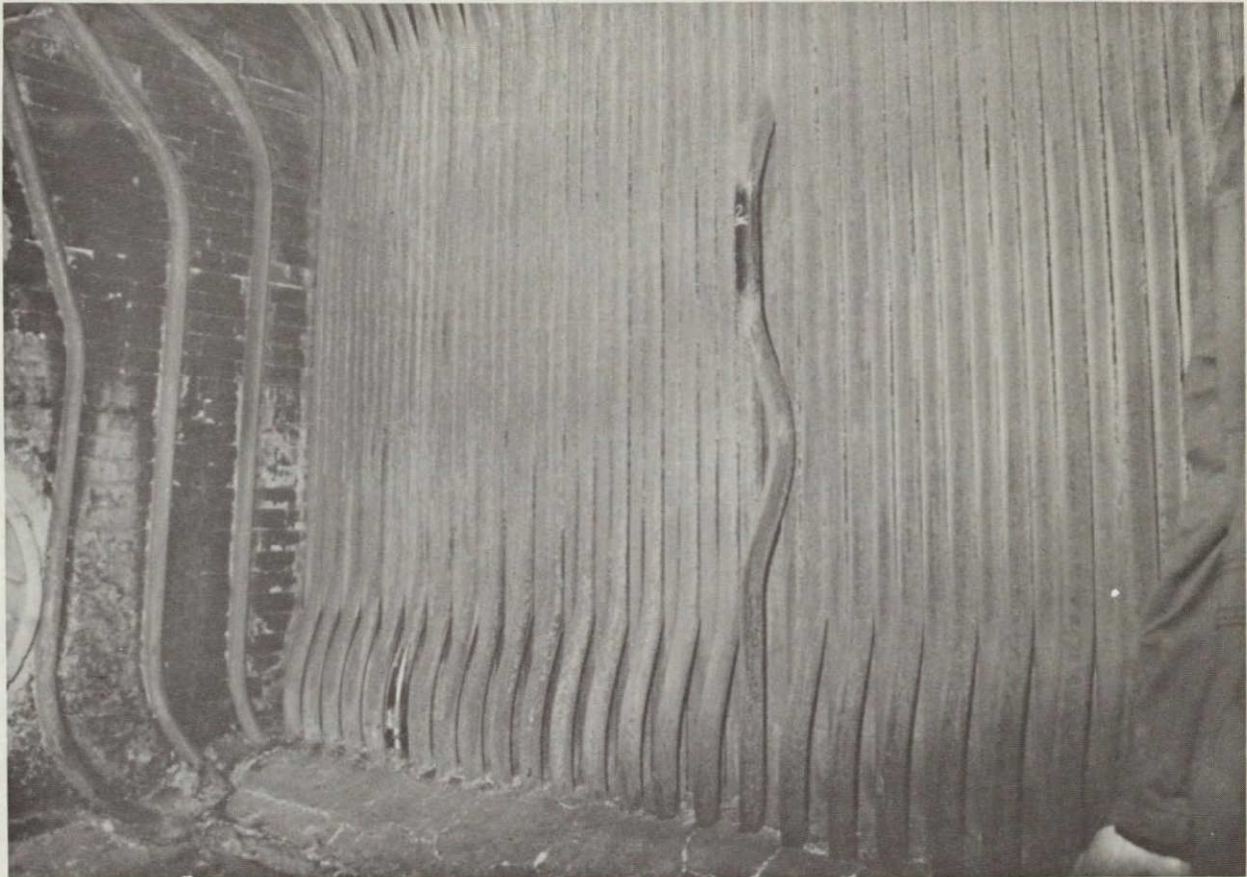
Figure 18A

## BOILER INTERIOR SCHEMATIC



Boiler Interior Schematic

Figure 18B

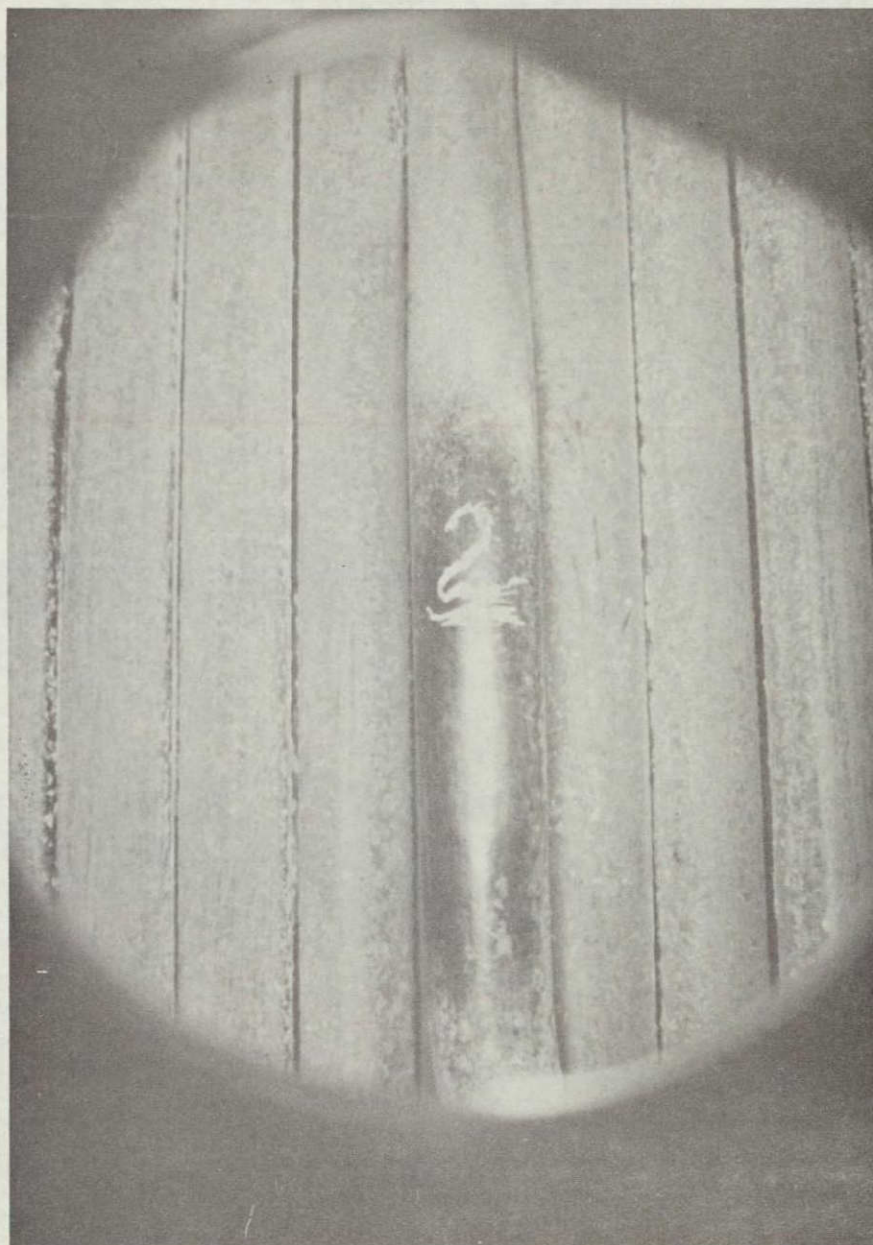


Clean Boiler Area - Including Tube 2

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Figure 18C



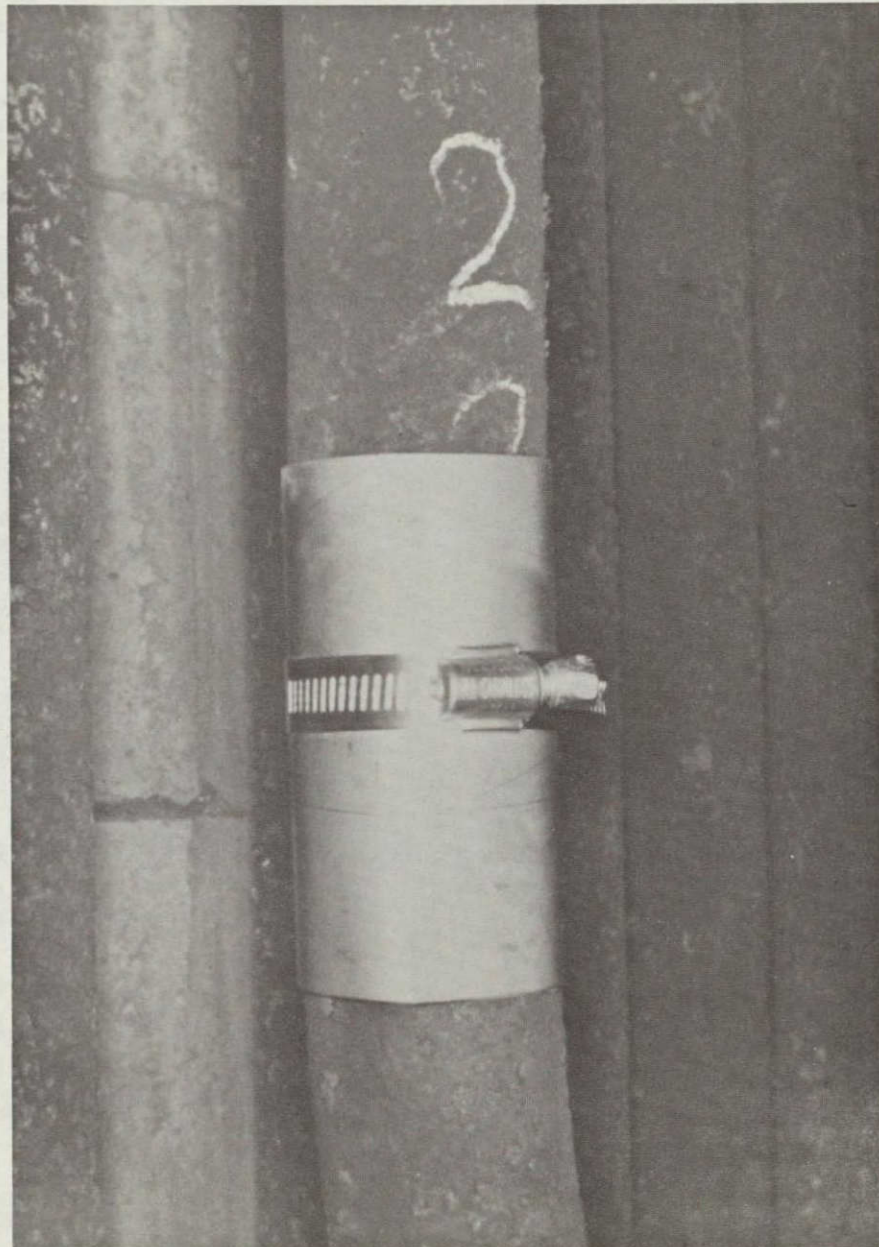
Boiler Test Tube #2 - Clean



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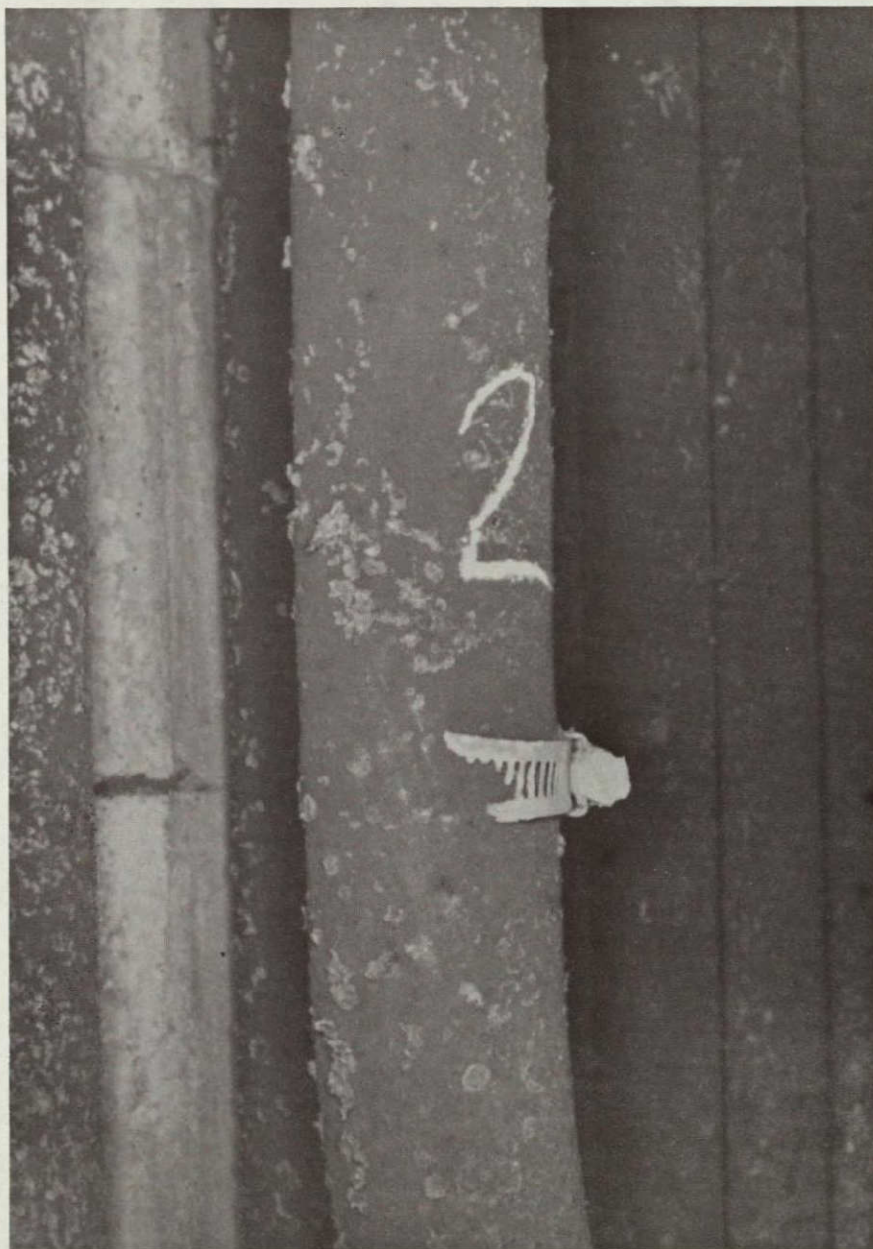
47.

Figure 18D



Boiler Test Tube #2 After Burning With  
Low Sulfur Oil

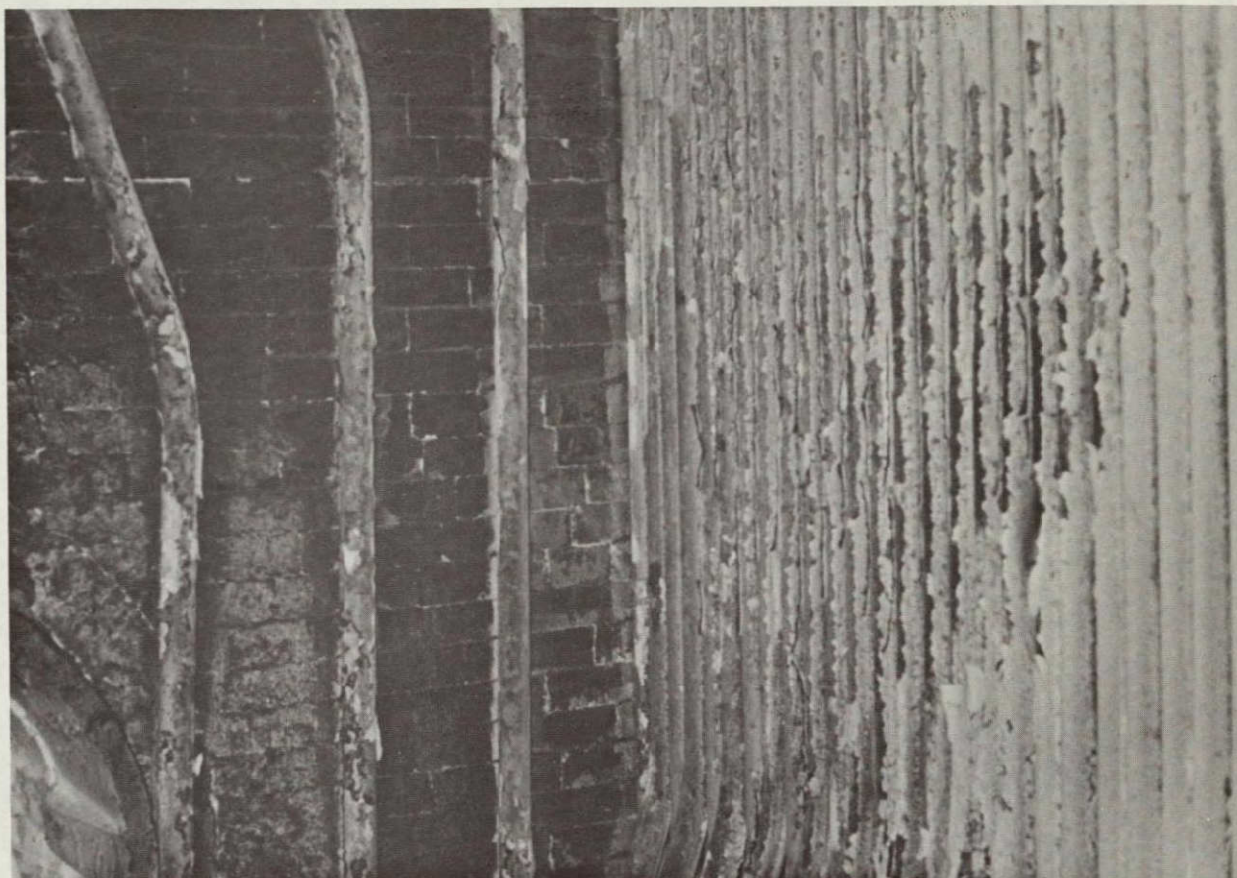
Figure 18E



Boiler Test Tube #2 After Burning With  
High Sulfur Oil



Figure 19A



Boiler Tube Wall After Run with Soda Ash

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Figure 19B



Boiler Tube #3 Before Start of Testing Runs



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Figure 19C

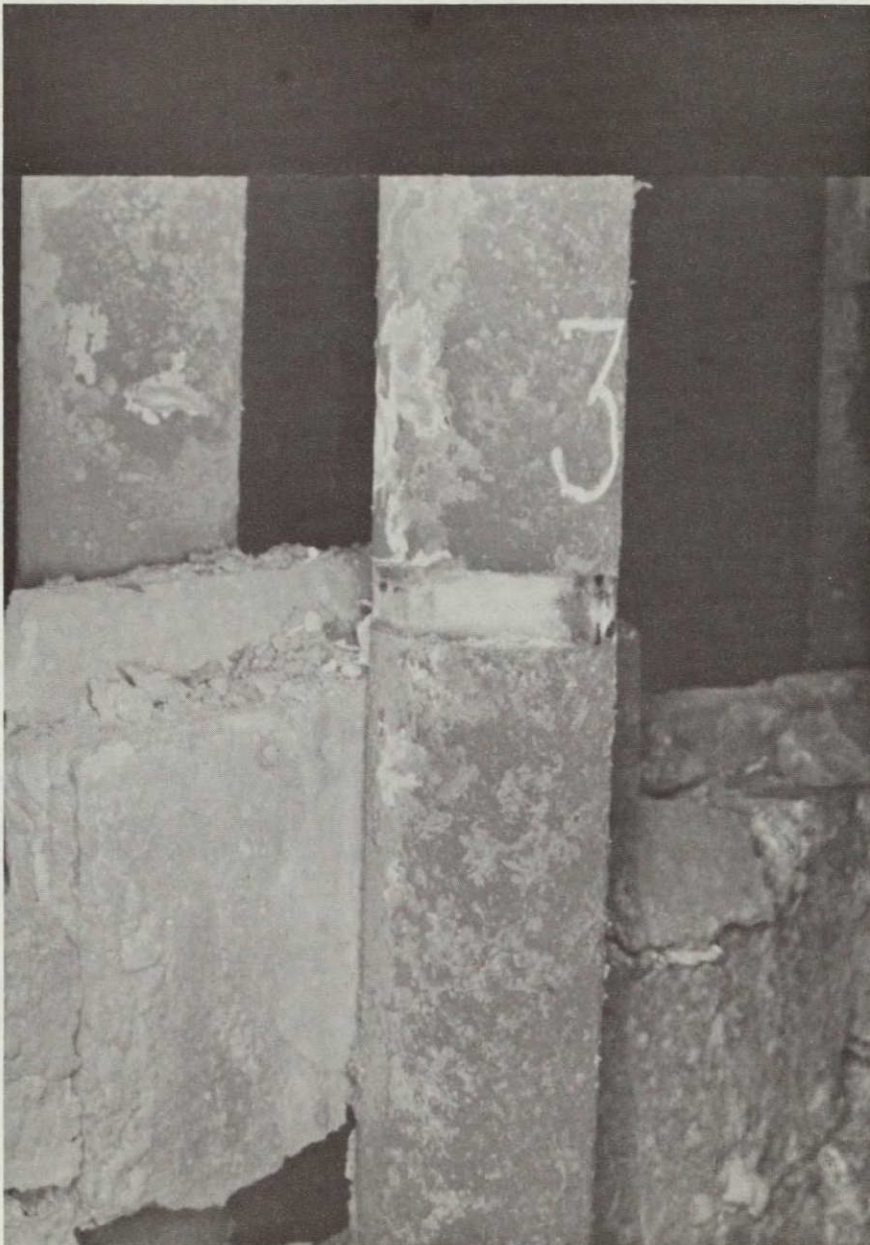


Test Tube #3 - After Run With Soda Ash



52.

Figure 19D

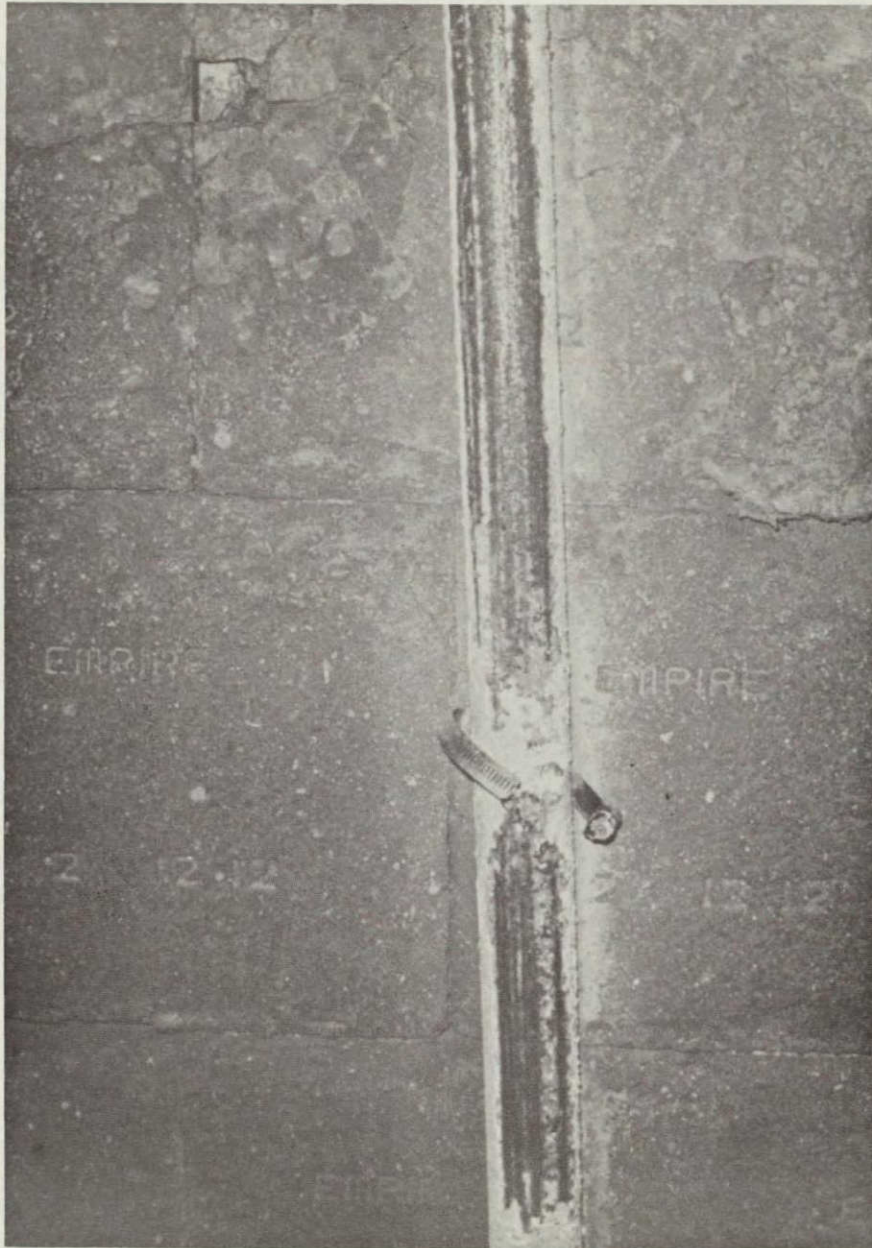


Test Tube #3 After Burning with Oil

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Figure 19E

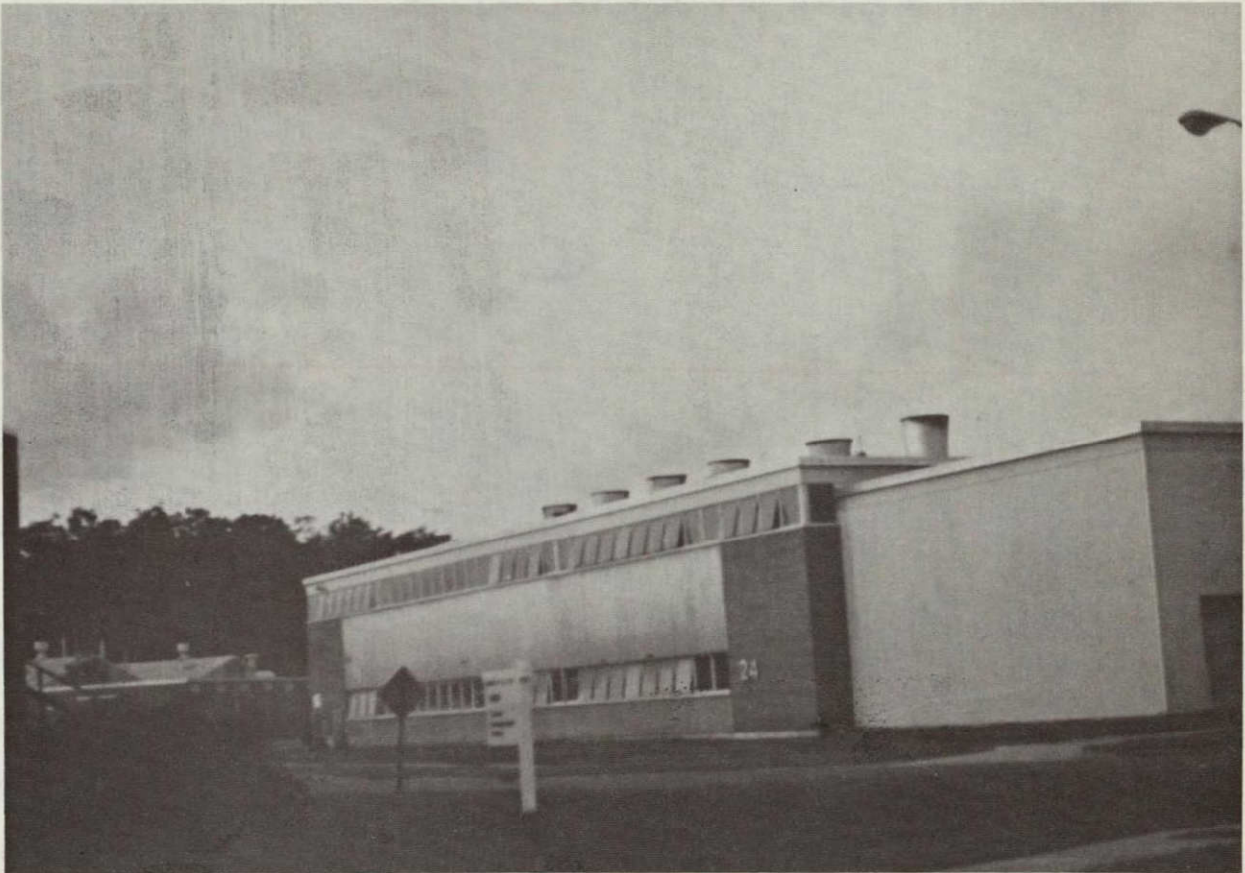


Test Tube Scraped Clean of Sodium Sulfate Particles



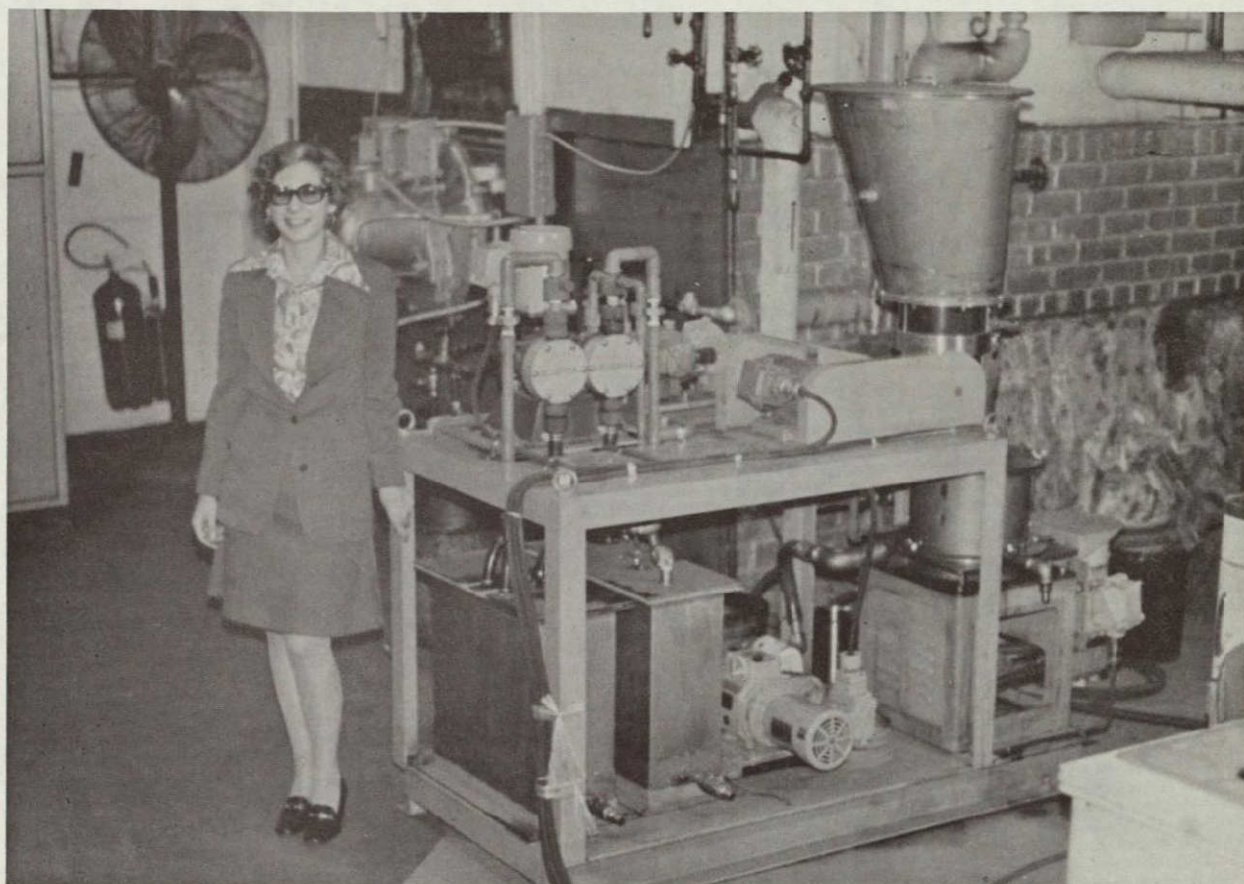
54.

Figure 19F



Boiler #1 Stack During Soda Ash Run

Figure 20A

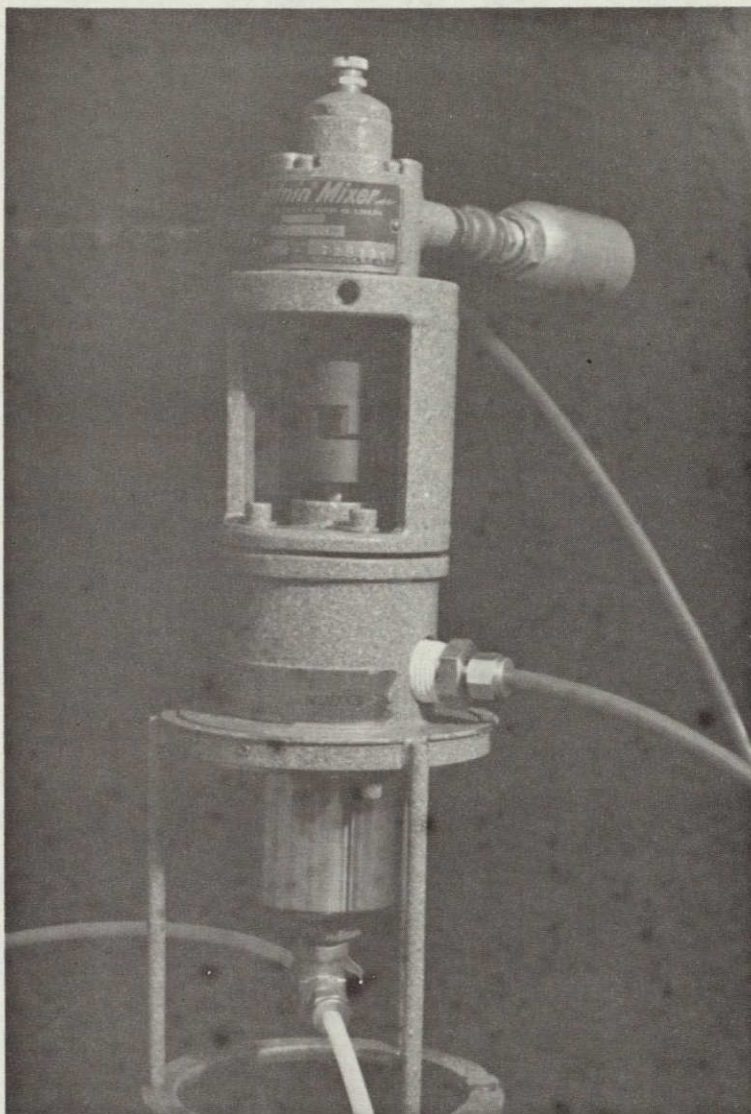


Funken Mixer In The  
Boiler Room at Adelphi University

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Figure 20B



Photograph of Lightin' Mixer

## APPENDIX B

HEAT LOSSES

The losses were calculated by the methods outlined in the ASME Power Test Code 4.1, and the total loss was determined from the equation.

$$L = L_G + L_H + L_{MA} + L_Z + L_E + L_{CO} + L_{UC} + L_{UHC} + L_B$$

where

- $L$  = total heat energy loss per pound of "as fired" fuel
- $L_G$  = heat loss carried away by dry flue gas per pound of A.F. fuel
- $L_H$  = heat loss due to moisture from the burning of hydrogen per pound of A.F. fuel
- $L_{MA}$  = heat loss due to moisture in the entering air per pound of A. F. fuel
- $L_Z$  = heat loss due to moisture in atomizing steam per pound of A. F. fuel
- $L_E$  = heat loss due to water added to fuel to generate emulsion per pound of A. F. fuel
- $L_{CO}$  = heat loss due to the incomplete combustion of the carbon per pound of A. F. fuel due to formation of carbon monoxide
- $L_{UC}$  = heat loss due to the soot (unburned carbon) per pound of A. F. fuel
- $L_{UHC}$  = heat loss due to unburned hydrocarbons per pound of A. F. fuel
- $L_B$  = heat loss due to radiation and convection

$L_G$  = heat energy carried away by dry flue gas

Now

$$L_G = W_G \times \bar{C}_P \times (t_2 - t_1)$$

where

$W_G$  = lbs. of dry gas/lb of A.F. fuel

$$W_G = \frac{44.01 (CO_2) + 32.00 (O_2) + 28.02 (N_2)}{12.01 (CO_2)} \left( C_b + \frac{12.01}{32.07} S \right)$$

with  $CO_2$ ,  $O_2$ , and  $N_2$  = percent by volume of dry flue gas,

$C_b$  = lbs. of carbon per lb. of "as fired" fuel,  $S$  = lb of sulfur per lb of "as fired" fuel

$\bar{C}_P$  = mean specific heat of the dry flue gas over the temperature range  $t_1 \leq t \leq t_2$

$t_2$  = stack or flue gas temperature

$t_1$  = intake air temperature (reference temperature)

$$\bar{C}_P = \frac{1}{100} \left[ (CO) \bar{C}_{p1} + (O) \bar{C}_{p2} + (N) \bar{C}_{p3} \right]$$

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with

$$\bar{C}_{p_1} = \frac{1}{t_2 - t_1} \int_{t_1}^{t_2} C_{p_1} dt$$

and similarly for  $\bar{C}_{p_2}$  and  $\bar{C}_{p_3}$  where

$$C_{p_1} = 16.2 - 6530 T^{-1} + 1,140,000 T^{-2} \text{ BTU/lb m/OR}$$

$$C_{p_2} = 11.515 - 172 T^{-1/2} + 1530 T^{-2} \text{ BTU/lb m/OR}$$

$$C_{p_3} = 9.47 - 3470 T^{-1} + 1,160,000 T^{-2} \text{ BTU/lb m/OR}$$

for  $\text{CO}_2$ ,  $\text{O}_2$  and  $\text{N}_2$ , respectively, with  $540^\circ\text{R} \leq t \leq 5,000^\circ\text{R}$   
 $(27^\circ\text{C} \leq t \leq 2,500^\circ\text{C})^{(1)}$

From these we find

$$\bar{C}_{p_1} = \frac{1}{t_2 - t_1} \left( 16.2 t - 6530 \ln t - \frac{1.41 \times 10^6}{t} \right) \Big|_{t_1}^{t_2}$$

$$\bar{C}_{p_2} = \frac{1}{t_2 - t_1} \left( 11.515 t - 344 t^{1/2} + 1530 \ln t \right) \Big|_{t_1}^{t_2}$$



$$\bar{C}_{p_3} = \frac{1}{t_2 - t_1} \left( 9.47 t - 3470 \ln t - \frac{1.16 \times 10^6}{t} \right) \Big|_{t_1}^{t_2}$$

From the measured values of  $t_1$  and  $t_2$  the values of  $\bar{C}_{p_1}$ ,  $\bar{C}_{p_2}$  and  $\bar{C}_{p_3}$  are readily determined, and the measured flue gas concentrations permits an evaluation of  $\bar{C}_p$  to be carried out.

$L_H$  = heat loss due to moisture from the burning of hydrogen

$$L_H = W_H \times H \times (h_t - h_r)$$

where

$$W_H = \frac{\text{lbs of water produced}}{\text{lb of hydrogen}} = 8.936$$

$$H = \frac{\text{lb. of hydrogen}}{\text{lb of A. F fuel}} \quad (\text{by analysis})$$

$$h_t = \text{enthalpy of water vapor at temperature } t_2$$

$$h_r = \text{enthalpy of water (liquid) at reference temperature } t_1$$

$h_t$  and  $h_r$  are found from the Steam Tables <sup>(2)</sup> to be

$$h_t = 1260 + 0.987 (t_2 - 235^\circ\text{C}) \quad \text{BTU/lb} \quad \pm 0.3 \text{ BTU/lb}$$

$$h_r = 45.1 + 1.796 (t - 25^\circ\text{C}) \quad \text{BTU/lb} \quad \pm 0.1 \text{ BTU/lb}$$

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$L_{MA}$  = heat loss due to moisture in entering air

$$= W_m \times W_d \times (h_t - h_{rv})$$

where

$W_m$  = lb. of moisture (water vapor)/lb of dry air

$W_d$  = lb. of dry air supplies/lb. of A.F. fuel

$h_t$  = enthalpy of water vapor at temperature  $t_z$

$h_{rv}$  = enthalpy of water vapor at reference temperature  $t_1$

An expression for  $h_t$  is given above while

$$h_{rv} = 1099 + 0.78 (t_1 - 30^\circ\text{C}) \text{ BTU/lb.}$$

and

$$W_d = \frac{1}{0.7685} \frac{28.02 (N_2)}{12.01 (CO_2)} \left( C_b + \frac{12.01}{32.07} S \right)$$

where the symbols  $N_2$ ,  $CO_2$ ,  $C_b$  and  $S$  have been defined above in this Appendix

$L_z$  = heat loss due to moisture supplied by atomizing steam

Since the atomizing steam was supplied from the output steam but beyond the orifice plate, the heat energy required to generate this steam is already included in the energy output. However, when returned as atomizing steam, further energy is supplied to this water vapor before it emerges as a component of the flue gas. Hence,

$$L_z = \frac{W_z}{W_f} (h_t - h_{rs})$$

where

$W_z$  = atomizing steam flow rate

$W_f$  = measured fuel flow

$h_t$  = enthalpy of water vapor at temperature  $t_2$   
(see above)

$h_{rs}$  = enthalpy of entering atomizing steam  
 $\approx 1187$  BTU/lb

$L_E$  = heat loss due to the water added to fuel to create emulsion

$$= \frac{W_w}{W_f} (h_t - h_r)$$

where

$W_w$  = water flow rate

$W_f$  = measured fuel flow

$h_t$  = enthalpy of water vapor at temperature  $t_2$   
(see above)

$h_r$  = enthalpy of water (liquid) at temperature  $t_1$

$L_{CO} \approx$  heat loss due to formation of carbon monoxide

$\approx 0$  as no measurable concentrations of CO were observed

$L_{UC}$  = heat loss due to soot formation  
(unburned carbon)

$$\cong \frac{W_c}{W_f} \times H_s + \frac{W_c}{W_f} \times C_s \times (t_2 - t_1)$$

where

$W_c$  = amount of soot generated per hour (lbs/hr)

$W_f$  = measured fuel rate in lbs/hr

$H_s$  = heat energy content of the soot

$C_s$  = specific heat capacity of the soot

$t_2$  = flue gas temperature, as above

$t_1$  = reference air temperature, as above

$L_{UHC}$  = heat loss due to unburned hydrocarbons

$\cong 0$  as no detectable amounts of hydrocarbons  
were present in the flue gasses

$L_B$  = heat loss due to radiation and convection

$\cong 1.0\%$  of heat energy input as determined by  
Figures 8 and 9 of the ASME Power Test Code  
PTC 4.1.

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## Fuel Analysis and Data of Test #35

Fuel.	high sulfur no. 6 oil
Fuel flow rate.	178.8 gal/hr
Steam pressure:	159 psia
Pressure difference across orifice plate:	1.700
Steam temperature.	184.6°C
Water temperature.	113.9°C
Air temperature ( $t_1$ )	34°C
Oil temperature.	84°C
Water temperature (for emulsion).	72°C
Water in emulsion	12%
Temperature of flue gas ( $t_2$ ).	254°C

## Fuel data

Carbon	83.71%
Hydrogen	10.67%
Sulfur	2.43%
Density	0.9488
BTU/lb	18,329
BTU/gal	145,040

## Flue gas analysis

CO <sub>2</sub>	12.18%
O <sub>2</sub>	4.27%
N <sub>2</sub>	83.55%
Excess Air	24%

Specific enthalpy of feed water	205.4 BTU/lb
Specific enthalpy of steam	1196.9 BTU/lb
Actual water evaporated.	20,630 lbs/hr

Total heat input:	$26.04 \times 10^6$	BTU/hr
Total heat output	$20.45 \times 10^6$	BTU/hr
Thermal efficiency	78.5%	

#### Sample Calculation of Losses

To determine the heat losses for test No.35 we first calculate  $L_G$ . For this we need  $W_G$  and  $\bar{C}_p$ . Since

$$W_G = \frac{44.01 (CO_2) + 32.00 (O_2) + 28.02 (N_2)}{12.01 (CO_2)} \left( C_b + \frac{12.01}{32.07} S \right)$$

and

$$\bar{C}_p = \frac{1}{100} \left[ \bar{C}_{p_1} (CO_2) + \bar{C}_{p_2} (O_2) + \bar{C}_{p_3} (N_2) \right]$$

and  $CO_2$ ,  $O_2$  and  $N_2$  are 12.18%, 4.27% and 83.55% respectively

According to analysis  $C_b = 0.8371$  lb of carbon/lb of A.F. fuel and  $S = 0.0243$  lb. of sulfur/lb of A.F. fuel.

From these data we find

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$$W_G = 20.60 \times 0.8462 = 17.43 \text{ lbs of dry flue gas per} \\ \text{lb. of A.F. fuel}$$

This data coupled with  $t_2 = 254^\circ\text{C}$  and  $t_1 = 34^\circ\text{C}$  yields

$$\bar{C}_p = 0.437 \text{ BTU/lb/}^\circ\text{C (or 0.243 BTU/lb/}^\circ\text{F)}$$

$$L_G = W_G \times \bar{C}_p \times (t_2 - t_1)$$

$$= (17.43) (0.437) (254-34)$$

$$L_G = 1676 \text{ BTU/lb of A.F. fuel } \pm 40 \text{ BTU/lb}$$

The uncertainty is due to the fluctuations in the concentrations, point-by-point monitoring at 15 minute intervals, instrument error, and 1% limitation on  $C_p$  data taken from reference.

$$L_H = \text{heat loss due to moisture from burning of hydrogen}$$

$$= 8936 H (h_t - h_r)$$

$$H = 0.1067 \text{ lb of hydrogen/lb. of A.F. fuel}$$

$$h_t = 1260 + 0.987 (t_2 - 235^\circ\text{C})$$

$$h_r = 45.1 + 1.796 (t_1 - 25^\circ\text{C})$$

$$t_2 = 254^\circ\text{C}$$

$$t_1 = 34^\circ\text{C}$$

$$h_t = 1278.8 \text{ BTU/lb}$$

$$h_r = 61.3 \text{ BTU/lb}$$

$$L_H = 1160.9 \text{ BTU/lb of A.F. fuel} \pm 8 \text{ BTU/lb}$$

The uncertainty in this figure is rather small and arises from error in the fuel analysis in the determination of percent hydrogen, variation in fuel from measured value, and/or temperature determinations

$$L_{MA} = \text{heat loss due to moisture in entering air}$$

At  $t_1 = 34^\circ\text{C}$  ( $93.2^\circ\text{F}$ ) the partial pressure of water vapor is 0.7 psia (saturated)

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$$\frac{n_1}{n_2} = \frac{\text{no. of moles of water vapor}}{\text{no. of moles of dry air}} = \frac{\text{partial pressure of water vapor}}{\text{partial pressure of dry air}}$$

$$= \frac{0.77}{13.93} \approx 0.055 \text{ moles of water vapor/mole of dry air}$$

$$W_m = \frac{\text{lb of moisture}}{\text{lb of dry air}} = \frac{0.055 \times 18 \text{ gm/mole of water}}{1.00 \times 29 \text{ g/mole of dry air}}$$

$$= 0.034 \text{ lb of H}_2\text{O per lb of dry air}$$

(if the air is saturated)

$$W_d = \text{lb of dry air supplied per lbs of fuel}$$

$$W_d = \frac{1}{0.7685} \frac{28.02 (N_2)}{12.01 (CO_2)} \left( C_b + \frac{12.01}{32.07} S \right)$$

Since, for this test No. 35,  $N_2 = 83.55\%$ ,  $CO_2 = 12.18\%$ ,  $C_b = 0.8371$  and  $S = 0.0243$ ,

$$W_d = \left( \frac{1}{0.7685} \right) (16.00) (0.8462)$$

$$W_d = 17.62 \text{ lbs of dry air/lb of fuel}$$

Since  $L_{MA} = W_m \times W_d \times (h_t - h_{rv})$ ,

$$L_{MA} = (0.034) (17.62) (1278.8 - 1102.1)$$

$$L_{MA} = 105.8 \text{ BTU/lb of A.F. fuel}$$

This calculation is based, as indicated, on air at a temperature of 34°C (93.2°F) and 100% relative humidity. However, a more reasonable estimate would be to assume a relative humidity for most tests of approximately 40%. Then for  $t_1 = 34^\circ\text{C}$  as for this test  $P_1 = (0.772 \text{ psia}) (0.40) = 0.309 \text{ psia}$  for water vapor and  $P_2 = 14.7 - 0.309 = 14.391 \text{ psia}$  for dry air; and

$$\frac{n_1}{n_2} = \frac{0.309}{14.391} = 0.0215 \text{ moles of water vapor per mole of dry air}$$

$$W_m = 0.033 \text{ lb of water vapor per lb of dry air}$$

In this case

$$L_{MA} = 41.4 \text{ BTU/lb of A F fuel}$$

Since most tests were conducted with an inlet air temperature ( $t_1$ ) of less than 34°C, and the relative humidity was not monitored at the inlet, a fixed value of

$$L_{MA} = 41 \text{ BTU/lb of A. F. fuel}$$

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was assigned to each test. Since  $L_{MA}$ , as in this case, could be as high as 105 BTU/lb, this corresponds to an uncertainty in the value of  $L_{MA}$  of 60 BTU/lb.

Further yet, in the last series of tests the inlet air temperature reached as high as 36°C corresponding to a pressure

at 100% relative humidity of 0.868 psia or a value of  $n_1 = 0.0627$  or  $W_m = 0.039$ . This yields a value for  $L_{MA}$  of  $L_{MA} = 121.4$  BTU/lb of A.F. fuel or an uncertainty in the value used of 80 BTU/lb or 0.4% of heat input.

$L_z$  = heat loss due to moisture supplied by atomizing  
steam

Even if  $W_z/W_f = 1$ , that is one pound of atomizing steam supplied per pound of fuel,

$$L_z = \frac{W_z}{W_f} (h_t - h_{rs})$$

$$L_z = 1 (1278.8 - 1187) = 92 \text{ BTU/lb of A.F. fuel}$$

The uncertainty in this figure may be as large as 50 BTU/lb or 0.27% of heat input.

$L_E$  = heat loss due to the water added to fuel to  
creat emulsion

In this test (18-21) the emulsion contained 12% water or  
 $W_w/W_f = 0.12$

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$$\begin{aligned}
 L_E &= \frac{W_w}{W_f} \times (h_t - h_r) \\
 &= 0.12 (1278.8 - 61.3) = 146 \text{ BTU/lb} \pm 15 \text{ BTU/lb}
 \end{aligned}$$

$L_B$  = heat loss due to radiation and convection

$$L_B \cong 1\% \text{ of heat input} = 0.01 \times 18,329$$

$$L_B \cong 183 \text{ BTU/lb of A. F. fuel}$$

For this test the total heat loss becomes

$$\begin{aligned}
 L &= L_G + L_H + L_{MA} + L_Z + L_E + L_B \\
 &= 1676 + 1161 + 41 + 92 \pm 146 + 183
 \end{aligned}$$

$$L = 3299 \pm 191 \text{ BTU/lb of A. F. fuel}$$

or in terms of energy input

$$L = 18.0 \pm 1.0\%$$

One source of error has been neglected in the previous discussions and that arises from water in the fuel. The fuel contains water but the analysis from Peniman and Browne was not precise on the amount, residues and water were listed together. In the early tests, where a blend of number 2 and number 6 oils were used, approximately 0.8% consisted of residues and water, while in the later tests, where a high sulfur oil was used, approximately 3% occurs as residues and water. This water appears as correction

to the BTU content of the oil where

$$H_{fv} = H_{f'} \frac{100 - m}{100}$$

and  $H_{f'}$  = BTU content of oil (dry basis) at constant volume,  
 $m$  = percent moisture

Finally, the BTU content at constant pressure for these oils is

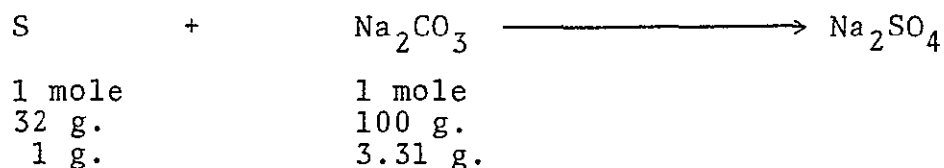
$$H_{fp} = H_{fv} + 29 \text{ BTU/lb}$$

Since this appears as a correction to the BTU content of the oil, this directly affects the calculation of the thermal efficiency by input/output method as well as the per cent losses. Until a more precise measurement of this water content is carried out we can only estimate this effect as approximately 1/2% in the first series of tests and 1-2% in both thermal efficiency and heat losses in the latter series of tests.

- (1) Young, Basic Engineering Thermodynamics, (pp 434)
- (2) Keenan, J.H., Keyes, F. G., Hill, P.G., and Moore, J. G.,  
Steam Tables, John Wiley & Sons, Inc., New York, 1969

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## APPENDIX C

SO<sub>2</sub> CALCULATION

1 gallon number 6 oil

$$3785 \text{ ml} \times 0.95 \text{ (density at } 70^\circ\text{C)} = 3595.75 \text{ g.}$$

$$2.43\% \text{ S of } 3595.75 = 87.38$$

$$87.38 \text{ g.} \times 3.31 = 289.23 \text{ g Na}_2\text{CO}_3 \text{ required}$$

For 20% Na<sub>2</sub>CO<sub>3</sub> & using 19% H<sub>2</sub>O (based on oil)

$$3785 \text{ g.} \times 0.19 \times 0.20 = 143.8 \text{ g. Na}_2\text{CO}_3 \text{ used.}$$

$$\frac{143.8 \text{ g.}}{289.23} = 50\% \text{ Stoichiometric amount used}$$


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SO<sub>2</sub> REDUCTION

SO<sub>2</sub> (used w/o soda ash) 1340 ppm

SO<sub>2</sub> (used with soda ash) 670 ppm

$$\text{Therefore, } \frac{1340-670}{1340} = 50\% \text{ reduction}$$

$$\text{Efficiency of removal of S} = \frac{50\%}{50\%} = 100\%$$


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SO<sub>2</sub> CONCLUSION

For 100 gallons of oil, 19 gallons of 20% aqueous soda ash was added. For oil containing 2.43% sulfur, this corresponds to 50% of stoichiometric amount used.

The average SO<sub>2</sub> concentration without soda ash was 1340 ppm.

The average SO<sub>2</sub> concentration with soda ash was 670 ppm.

Therefore 50% reduction in SO<sub>2</sub> concentration.

$$\text{Efficiency of SO}_2 \text{ removal} = \frac{50\%}{50\%} = 100\%.$$

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